TRANSMISSION OF POWER

BY

FLUID PRESSURE.

AIR AND WATER.

BY

WILLIAM DONALDSON,

M.A., M. INST. C.E.;

AUTHOR OF 'TREATISE ON OBLIQUE ARCHES';
'SWITCHES AND CROSSINGS'; 'TABLES FOR PLATELAYERS'; 'SOLID BEAMS AND GIRDERS';
'PRINCIPLES OF CONSTRUCTION AND EFFICIENCY OF WATERWHEELS';
'PONCELET TURBINE'; AND 'WATER-PRESSURE ENGINE.'

E. & F. N. SPON, 125, STRAND, LONDON.

NEW YORK: 12, CORTLANDT STREET.

1888.

UP.

INTRODUCTION.

It is clear that an incompressible fluid like water, subject to no changes except freezing, which can easily be guarded against, must be a much better medium for transmitting power than an elastic fluid like air, which cannot be compressed without great increments of temperature corresponding to the increments of pressure. The absolute energy imparted to the air during compression is equal to the equivalent in work of the number of thermal units required to raise the temperature of the air to that due to adiabatic compression, and must therefore be wholly lost, if the air without doing work is cooled down to the original temperature of the free air. The work done by expansion down to atmospheric pressure after cooling corresponds to an equal loss of the absolute energy possessed by the air before compression. This cooling-down to the temperature of the medium surrounding the pipes is inevitable, when the air is transmitted to considerable distances, unless the pipes are coated with some non-conducting substance. Heat must also be lost in the very act of compressing the air.

The whole work done in the cylinder of the air-pump during one full stroke consists of two operations.

(1) Work done in raising the tension of the air to the required tension.

(2) Work done in forcing the compressed air into a receiver.

In the following pages the work done in raising the tension of the air is called the work of compression, and the sum of the two items (1) and (2) the work of pumping.

In nearly all the papers written in advocacy of compressed air for the transmission of power in preference to water, there occur the following fallacies which will be fully discussed hereafter. It has been stated—

(1) That there is no appreciable loss of energy due to pipe friction. This conclusion is based on two errors, viz. the confounding of the absolute energy of compressed air with the sensible energy, i. e. the energy available for working motors, and the belief that the rise in the temperature of the compressed air due to friction will practically compensate for the work done in overcoming friction.

(2) That the energy lost by cooling can be restored by reheating, without entailing a still heavier loss.

(3) That owing to the shock caused by the sudden changes in velocity in a heavy fluid like water, only small velocities are permissible. *Instantaneous* changes can never occur even where a single machine only is operative, and the bursting force of the shocks can always be guarded against by using a well-charged air-vessel. This opinion, however, seems to be based on a misconception, viz. that the bursting effect of the shocks varies as the change of *vis viva.* In reality it varies as the change of momentum.

(4) That greater power can be transmitted by air

compressed up to 45 lbs. pressure than by water up to 800 lbs. The contrary is the case. This opinion is based on the assumption that the actual loss by pipe friction must necessarily be the same in both cases. In discussing questions of efficiency, however, we have only to deal with percentages. Thus, if we fix the limit of loss in transmission at five per cent. of the maximum pressure in each case, the actual loss in the case of air at 45 lbs. pressure must not exceed 2¼ lbs., whilst in that of the water at 800 lbs. pressure, it may be 40 lbs.

CONTENTS.

TRANSMISSION OF POWER

BY

FLUID PRESSURE.

AIR AND WATER.

CHAPTER I.

EFFICIENCY OF COMPRESSED AIR.

Absolute and indicated fluid pressure.

ALL bodies whether solid or fluid are subjected to the pressure of the atmosphere. The mean pressure of the atmosphere at the level of the sea is 14·7 lbs. per square inch, and is equivalent to the weight of (*a*) a column of air at 32° F., 27,801 feet, or about 5¼ miles high, of uniform density, equal to that of the air at mean sea-level; (*b*) a column of mercury 29·922 inches high at 32° Fahr., and 30 inches high at 62° F.; (*c*) a column of water at 62° F. 33·947 feet, or nearly 34 feet high. When the pressures, to which fluids inclosed in a vessel are subjected, are ascertained by means of an ordinary pressure gauge, the reading of the gauge shows the difference between the total pressure and the pressure of the atmosphere at the time and place where the observation is made. Hence in order to ascertain the total pressure to which the fluid is subjected, it is necessary to add to the reading of the pressure gauge the reading of the barometer. The reading shown by the pressure gauge is called the *indicated*, and the total pressure the *absolute* pressure.

In analytical investigations the pressures are usually estimated in atmospheres, the equivalent of one atmosphere being a pressure of 14·7 lbs. to the square inch.

Absolute and indicated temperatures.

The volume of air and all perfect gases subjected to a constant pressure is found to be increased by as nearly as possible $\frac{1}{461}$ part of its volume for each degree Fahrenheit of increase of temperature, so that, if a thermometric scale be adopted in which the zero point of the Fahrenheit scale corresponds with 461°, the volume of any given weight of a perfect gas at constant pressure will vary as the temperature indicated on the thermometric scale, whose zero point will be 461° below the zero point of the Fahrenheit thermometer. This scale has been called the *absolute thermometric scale*, and the temperatures measured by it the *absolute* temperatures.

The temperatures shown by ordinary thermometers are called the *indicated* temperatures.

In the Fahrenheit scale, the freezing-point, the temperature of melting ice, is marked 32°, and the boiling-point of water under one atmosphere of absolute pressure 212°, so that there are 180° F. between the freezing and boiling points. In the Centigrade scale the freezing-point is marked zero, the boiling-point 100°; and in the Réaumur scale the freezing-point is marked zero, the boiling-point 80°. These scales are usually denoted by the initial capital letter of their names, so that

$$180^\circ \text{ F.} = 100^\circ \text{ C.} = 80^\circ \text{ R.}$$

The absolute temperatures of the freezing-points on the three scales will therefore be 493° F., 274° C., and 219° R., respectively.

When no other scale is hereafter specially mentioned, it must be understood that the Fahrenheit scale is always referred to.

Relations between the

If p, t be the absolute pressure and absolute temperature corresponding to the volume v of a given weight of air or any perfect gas, then, whatever may be the changes in the values of the three variables p, t, v, they have been found to be, by careful experimental research, always connected by the following equation of relation :—

absolute pressure, absolute temperature, and volume of a given weight of air under all conditions.

$$p\,v = a\,t,$$

where a is a constant. This relation is called the law of Boyle and Marriott. When the changes in the relative values of the three variables are brought about by mechanical means in such a way that nothing is added to or taken from the total heat, which is equal to the sum of the sensible and latent heat, the changes are said to be effected *adiabatically*. When the temperature is kept constant by the addition or subtraction of heat the changes are said to be effected *isothermally*.

Adiabatic and isothermal changes in the values of p, v, and t.

Specific heat.

The heat requisite to raise the temperature of one lb. of water from 32° F. to 33° F. has been adopted as the thermal unit, and has been called the *specific heat* of water at 32° F. The number of such units required to raise by 1° F. the temperature of any other body in any assigned condition, whether solid, liquid, or gaseous, is called the specific heat of that body under those conditions.

Specific heat of gases at constant volume less than specific heat at constant pressure.

In the case of gases the heat required to raise the temperature of a given weight of gas by one degree F. at constant volume and varying pressure is less than the heat required to raise the temperature at constant pressure and varying volume, because the gas in expanding does work which is accompanied by an equivalent loss of sensible heat. In the case of air the values of both kinds of specific heat are nearly constant for all temperatures, and are as follows :—

Specific heat at constant volume	·1688
„ „ pressure ..	·2377

Mechanical equivalent of one thermal unit.

According to the result of Joule's experiments, the approximate correctness of which has been confirmed by the researches of other scientific men, the number of foot-lbs. of work required to raise the temperature of one lb. of water at 32° F. one degree is 772 foot-lbs. This then is the approximate mechanical equivalent of one thermal unit. In honour of Joule it is denoted by the capital letter J.

Relations between the variables p, t, v when the changes are effected adiabatically.

Since experimental researches have proved that the relation

$$p\,v = a\,t$$

holds true for all values of the variables, whether the changes are brought about adiabatically or isothermally, the following relation

$$p\,d\,v + v\,d\,p = a\,d\,t \qquad (a)$$

must also hold true under all conditions when all the quantities vary. Also when v and t only vary,

$$p\,d\,v = a\,d\,t,$$

and when p and t only vary

$$v\,d\,p = a\,d\,t.$$

If c, k, represent the specific heat at constant pressure and constant volume respectively, then the heat required to raise the temperature of one lb. of the gas $d\,t$ degrees at constant pressure will be $c\,d\,t = \frac{c\,p\,d\,v}{a}$, and at constant volume $k\,d\,t = \frac{k\,v\,d\,p}{a}$. Since in adiabatic compression and expansion no heat is either imparted or abstracted, the sum of these two must under all adiabatic conditions be zero, and the following adiabatic relations between the variables p and v must always exist,

$$c\,p\,d\,v + k\,v\,d\,p = 0. \qquad (b)$$

If we substitute in (a) for $p\,d\,v$ its value from (b), viz. $-\dfrac{k\,v\,d\,p}{c}$, we shall get the relation

$$\frac{c-k}{c}\,.\,v\,d\,p = a\,d\,t\,;$$

but

$$v = \frac{a\,t}{p},$$

therefore

$$\frac{c-k}{c}\cdot\frac{d\,p}{p} = \frac{d\,t}{t}.$$

Integrating we get

$$\log_e p^{\frac{\gamma-1}{\gamma}} = \log_e m\,t$$

where m is a constant and

$$\gamma = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$$

$$= \frac{c}{k} = \frac{\cdot 2377}{\cdot 1688} = 1\cdot 408.$$

The value of this ratio has been determined in different ways. Solely, as above, by ascertaining experimentally the values of c and k and by calculations based on the results of experiments made to ascertain the velocity of light and sound. Many authorities prefer $1\cdot 401$ for the value of γ.

Equation (c) is equivalent to

$$p^{\frac{\gamma-1}{\gamma}} = m\,t,$$

and if, therefore, $p_0\,v_0\,t_0$ be other adiabatic values of the variables, we have

$$t = \left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}} t_0\,.$$

Also, since

$$p\,v = \frac{p_0\,v_0\,t}{t_0} = p_0\,v_0\left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}}$$

$$p\,v^\gamma = p_0\,v_0{}^\gamma = \text{a constant.}$$

Absolute and sensible energy of compressed air.

The *absolute energy* of a given weight of compressed air is equal to the energy that would be given out in expanding against pressure, whilst its temperature sinks to the zero point of the absolute thermometric scale. So long as the temperature of a given weight of air remains constant, the *absolute* energy is the same for all pressures.

In expanding adiabatically to any absolute pressure p_0 greater than zero from any pressures p_1, p_2, the energy given out in the two cases will be in the ratio of the fall of temperature. If T be the initial temperature common to the initial pressures p_1, p_2, the final temperatures will be $T\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$ and $T\left(\frac{p_0}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$ respectively, and the energy given out in the two cases will be in the ratio

$$\frac{1-\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}}{1-\left(\frac{p_0}{p_2}\right)^{\frac{\gamma-1}{\gamma}}};$$

the more nearly the final absolute pressure approaches the value zero, the more nearly will this ratio approach to its ultimate value, which is one of equality.

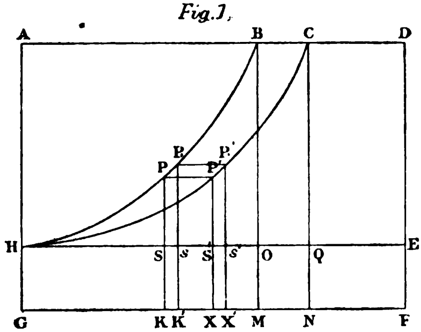
In actual work, unless means are provided for producing a vacuum, the pressure p_0 cannot fall below that of the atmosphere, and the energy given out during the fall to this pressure may be called the *sensible energy* of the compressed air.

The author has not met with the terms absolute and sensible energy in works on the use of compressed air, but their utility is at once evident.

Graphic illustration of the work done in compression.

In the diagram Fig. 1, let G H represent the initial absolute pressure before, and G A after compression, F G the original volume, F M the volume of compressed air at the temperature due to adiabatic compression, and F N its volume after having been cooled down to the initial temperature.

Fig. 1.



Complete the parallelogram A D F G and draw H E parallel to G F. Through M N draw M O B, N Q C meeting the line of initial absolute pressure H E in the points O and Q respectively, and the line of maximum pressure A D in the points B and C respectively. Then will N Q = M O = G H represent the initial, and N C = M B = G A the final absolute pressure. Draw the adiabatic curve H P B and the isothermal curve H P C. Then will the area B P H G M represent the whole work done in adiabatic compression, and the area C P′ H G N in isothermal compression. Since the work done by the air at the initial pressure is represented by the area H G M O in adiabatic, and H G N Q in isothermal compression, the net work done by the compressing engine is represented by the area B P H O in adiabatic, and by the area C P′ H Q in isothermal compression.

Similarly the whole work done after adiabatic compression in forcing the compressed air into the receiver is represented by the area B M F D and after isothermal by C N F D. Since the work done by the air in the corresponding periods is represented by the areas O M F E and Q N F E, the net work done by the engine in forcing the air into the receiver will be represented by the area B O E D after adiabatic, and by the area C Q E D after isothermal compression.

The work done by the engine in adiabatic pumping is represented by the area B P H E D, and in isothermal by the area C P′ H E D. The work done by the air in pumping is the same in both cases, and is represented by the area H G F E.

Work done in adiabatic compression.

Let $p_0 v_0 t_0$, $p_1 v_1 t_1$ be the absolute pressures, volumes, and absolute temperatures of any given weight of air before and after compression from pressure p_0 to p_1 and $p\,v\,t$ the corresponding values at any point P in the curve of compression. Let P_1 be a point contiguous to P, at which the pressure is $p + dp$. Draw P S K, $P_1\,s$ K′ cutting the line H E of initial absolute pressure in S and s and the line of volume G F in K and K′ respectively, then will G K, G K′ represent the volume swept out by the piston of the compressor, whilst the pressure is being increased from p_0 to p and $p + dp$ respectively. If these volumes be represented by V and V + dV respectively, the whole work done in raising the pressure of air from p to $p + dp$ will lie between $p\,d\mathrm{V}$ and $(p + dp)\,d\mathrm{V}$, and when dp, $d\mathrm{V}$ are indefinitely diminished, will be equal to $p\,d\mathrm{V}$, and the whole work done in raising the pressure from p_0 to p_1 to the integral

$$\int_{p_0}^{p_1} p\,d\mathrm{V}.$$

Now

$$p\,(v_0 - \mathrm{V})^{\gamma} = p\,v^{\gamma} = \text{constant};$$

therefore

$$(v_0 - \mathrm{V})^{\gamma}\,d\,p - \gamma p\,(v_0 - \mathrm{V})^{\gamma - 1}\,d\,\mathrm{V} = 0;$$

also,

$$p\,(v_0 - \mathrm{V}) = p\,v = \frac{p_0\,v_0\,t}{t_0} = p_0\,v_0\left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}};$$

therefore

$$p\,d\,\mathrm{V} = \frac{(v_0 - \mathrm{V})\,d\,p}{\gamma} = \frac{v_0\,p_0^{\frac{1}{\gamma}}\,d\,p}{\gamma\,p^{\frac{1}{\gamma}}}$$

and therefore

$$\int_{p_0}^{p_1} p\,d\,\mathrm{V} = \frac{v_0\,p_0^{\frac{1}{\gamma}}}{\gamma}\int_{p_0}^{p_1}\frac{d\,p}{p^{\frac{1}{\gamma}}}$$

$$= \frac{p_0\,v_0}{\gamma - 1}\left(\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \qquad \text{I.}$$

$$= \frac{p_1\,v_1}{\gamma - 1}\left(1 - \left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right) \qquad \text{II.}$$

$$= \frac{p_1\,v_1 - p_0\,v_0}{\gamma - 1}. \qquad \text{III.}$$

The symbol R will in future be used to denote the ratio $p_1 \div p_0$ of the final to the initial absolute pressure.

The product $p_0\,v_0$ is equal to the area H G F E, and therefore to the work done by the air in *pumping.* The product $p_1\,v_1$ is equal to the area B M F D, and therefore to the whole work of forcing the compressed air into the receiver.

Total work done in adiabatic pumping.

The total work done in pumping is equal to the whole work done in compressing, which is given by each of the equations I., II., and III., plus $p_1\,v_1$, the whole work done in forcing the compressed air into the receiver, or to

$$\frac{p_1\,v_1 - p_0\,v_0}{\gamma - 1} + p_1\,v_1.$$

Work done by the engine in adiabatic pumping.

The work done by the engine is equal to the whole work done during adiabatic pumping, less the work done by the air, viz. $p_0 v_0$, and is therefore equal to

$$\frac{p_1 v_1 - p_0 v_0}{\gamma - 1} + p_1 v_1 - p_0 v_0 = \gamma \left(\frac{p_1 v_1 - p_0 v_0}{\gamma - 1}\right) \quad \text{IV.}$$

so that the net work done by the engine in pumping is equal to γ times the whole work done in compression.

Work done by the engine in adiabatic compression.

The work done by the engine in adiabatic compression is equal to the whole work less that done by the air represented by the area H G M O, which is equal to

$$\text{H G . G M} = \text{H G (G F} - \text{F M)};$$

and

$$\text{F M} = \frac{\text{F N}.t}{t_0}.$$

Now F N the volume under isothermal compression is equal to $\frac{p_0 v_0}{p_1}$, and therefore the work done by the air in adiabatic compression is equal to

$$p_0 \left(v_0 - \frac{p_0 v_0}{p_1}\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}}\right) = p_0 v_0 \left(1 - \left(\frac{p_0}{p_1}\right)^{\frac{1}{\gamma}}\right),$$

and therefore the work done by the engine is equal to

$$\frac{p_0 v_0 \left(R^{\frac{\gamma-1}{\gamma}} - 1\right)}{\gamma - 1} - p_0 v_0 \left(1 - \frac{1}{R^{\frac{1}{\gamma}}}\right),$$

or to

$$\frac{p_0 v_0 \left(R^{\frac{\gamma-1}{\gamma}} + \frac{\gamma - 1}{R^{\frac{1}{\gamma}}} - \gamma\right)}{\gamma - 1}, \quad \text{V.}$$

and the ratio of this to the whole work done in compression will be therefore

$$\frac{R^{\frac{\gamma-1}{\gamma}} + \dfrac{\gamma - 1}{R^{\frac{1}{\gamma}}} - \gamma}{R^{\frac{\gamma-1}{\gamma}} - 1}. \qquad \text{VI.}$$

Since the work done by the engine in pumping is equal to γ times the whole work done in compression, the ratio of the work done by the engine in compression to the work done by the engine in pumping will be equal to the above ratio divided by γ.

Through P and P_1, Fig. 1, draw $P P'$ and $P_1 P_1'$ parallel to G F, meeting the isothermal curve in the points P' and P_1', then will the pressures at P' and P_1' be equal to the pressures at the points P and P_1 in the adiabatic curve, and the work done in raising the pressure of the air from p to $p + d\,p$ ultimately equal to $p\,d\,p$, and the whole work in raising the pressure from p_0 to p_1 to the integral

Work done in isothermal compression.

$$\int_{p_0}^{p_1} p\,d\,V$$

if V, V + d V represent the volumes G X, G X'.

Now in isothermal compression

$$p\,(v_0 - V) = p\,v = p_0\,v_0 = \text{constant}.$$
$$(v_0 - V)\,d\,p - p\,d\,V = 0.$$

Therefore
$$p\,d\,V = \frac{p_0\,v_0\,d\,p}{p},$$

and the whole work done in compression will be equal to

$$p_0\,v_0 \log_e \left(\frac{p_1}{p_0}\right) = p_0\,v_0 \log_e R. \qquad \text{VI.}$$

The whole work done in pumping isothermally is equal to the work done in compressing plus the work

Whole work done in pumping isothermally.

done in forcing the compressed air into the receiver, viz. $p_1 v_1$. Since the temperature is always the same, the product $p v$ is constant for all values of p and v, and therefore $p_1 v_1 = p_0 v_0$, and the whole work done is equal to

$$p_0 v_0 \log_e R + p_0 v_0 .$$

Work done by the engine in isothermal pumping.

Since $p_0 v_0$ expresses the work done by the air, the work done by the engine is equal to

$$p_0 v_0 \log_e R, \qquad \text{VII.}$$

or to the whole work done in compression.

Work done by the engine during compression.

The work done by the air during compression is equal to the area H G N Q, and is therefore equal to

$$HG \,.\, (GF - FN) = p_0 \left(v_0 - \frac{p_0 v_0}{p_1}\right)$$
$$= p_0 v_0 \left(1 - \frac{1}{R}\right),$$

and therefore the work done by the engine is equal to

$$p_0 v_0 \left(\log_e R + \frac{1}{R} - 1\right),$$

and the ratio of this to the whole work done in compression and to the net work done by the engine in pumping is

$$\frac{\left(\log_e R + \frac{1}{R} - 1\right)}{\log_e R}. \qquad \text{VIII.}$$

Comparison of the work done by the engine in adiabatic and isothermal compression and pumping.

The work done by the engine in adiabatic compression is equal to

$$\frac{p_0 v_0 \left(R^{\frac{\gamma - 1}{\gamma}} + \frac{\gamma - 1}{R^{\frac{1}{\gamma}}} - \gamma \right)}{\gamma - 1},$$

and in isothermal compression to

$$p_0 v_0 \left(\log_e \mathrm{R} + \frac{1}{\mathrm{R}} - 1\right),$$

so that the ratio of the latter to the former is equal to

$$\frac{(\gamma - 1)\left(\log_e \mathrm{R} + \frac{1}{\mathrm{R}} - 1\right)}{\mathrm{R}^{\frac{\gamma-1}{\gamma}} + \frac{\gamma - 1}{\mathrm{R}^{\frac{1}{\gamma}}} - \gamma}.$$

The work done by the engine in adiabatic pumping is equal to

$$\frac{p_0 v_0 \gamma \left(\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1\right)}{\gamma - 1},$$

and in isothermal to

$$p_0 v_0 \log_e \mathrm{R},$$

so that the ratio of the latter to the former is

$$\frac{(\gamma - 1)\log_e \mathrm{R}}{\gamma\left(\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1\right)}.$$

The work done in air pumping is independent of the initial temperature, and depends solely on the initial pressures.

Since the volume v_0 is equal to the contents of the air-compressing cylinder, it has the same values for all temperatures, and therefore so long as the initial pressure p_0 remains unchanged the product $p_0 v_0$ is constant. Since the work done in compression is equal to

$$p_0 v_0 \left(\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1\right) \text{ adiabatically,}$$

and

$$p_0 v_0 \log_e \mathrm{R} \text{ isothermally,}$$

so long as the ratio R remains constant the work done in compression has the same value for all initial temperatures. Also since the work done in forcing the compressed air into the receivers is equal to

$$p_1 v_1 = p_0 v_0 \mathrm{R}^{\frac{\gamma-1}{\gamma}} \text{ adiabatically,}$$

and to $p_0 v_0$, isothermally, the value of this work is also independent of the initial temperature, so that the whole work of air-pumping is independent of the initial temperature.

Effect of initial heating of the air by contact with the hot metal.

Since the weight of the initial volume v_0 of the air varies inversely as its absolute temperature, if the initial temperature t_0 be increased by t, the weight of the compressed air will be to its weight, if the initial temperature had remained unchanged before compression, in the ratio of $\frac{t_0}{t + t_0}$. Now both the sensible and absolute energy of air at constant pressure and temperature vary directly as its weight, and therefore the value of the energy after compression calculated on the basis of the final temperature $t + t_0$ being maintained must be multiplied by $\frac{t_0}{t_1 + t_0}$ in order to ascertain its energy after cooling to t_0.

If we put $t = \frac{t_1 - t_0}{m}$ we get

$$\frac{t_0}{t_1 + t_0} = \frac{m\, t_0}{t_1 + (m - 1)\, t_0} = \frac{m}{R^{\frac{\gamma - 1}{\gamma}} + m - 1}.$$

The value of m cannot be determined experimentally on account of the rapid rate of compression, but it is not probable that the rise in initial temperature will be less than one-fourth of the rise due to adiabatic compression.

The whole work done in perfect adiabatic and isothermal compression the same whether done in one or several stages.

In order to simplify the proof we may suppose the rate of compression to be the same at each stage, since if true in this case, it will be true in all cases. If, therefore, n be the number of the stages of compression, and R the ratio of compression at each stage, R^n will be the final ratio of compression. If then $p_1 v_1$, $p_2 v_2$, &c., $p_{n-1} v_{n-1}$, $p_n v_n$ represent the products of the volumes multipled by the pressures after the 1st, 2nd, &c., n – 1st and nth compressions, the whole work done *adiabatically* on the n compressions will be equal to the sum of the work of the several compressions, or to the following divided by $\gamma - 1$.

$$\begin{array}{lr} & p_1 v_1 - p_0 v_0 \\ & p_2 v_2 - p_1 v_1, \text{\&c.} \\ & p_{n-1} v_{n-1} - p_{n-2} v_{n-2} \\ & p_n v_n - p_{n-1} v_{n-1} \\ \hline \text{Total} & p_n v_n - p_0 v_0. \end{array}$$

Now $\frac{p_n v_n - p_0 v_0}{\gamma - 1}$ is equal to the whole work done adiabatically in one compression, and therefore the work done in adiabatic compression depends solely on the ratio of the final to the initial pressure, and is independent of the number of stages.

In isothermal compression the work done at each stage will be equal to $p_0 v_0 \log_e \text{R}$ since

$$p_0 v_0 = p_1 v_1 = \text{\&c.} = p_{n-1} v_{n-1} = p_n v_n,$$

and therefore the whole of the work done in n stages will be equal to

$$n p_0 v_0 \log_e \text{R} = p_0 v_0 \log_e \text{R}^n,$$

or to the work done in a single stage. Since in the case of adiabatic pumping the net work done by the engine is equal to γ times, and in that of isothermal compression exactly equal to, the whole work done in compression in each case, the net work done by the engine in pumping by any number of stages is equal to the net work done in pumping by one stage.

Net work done in pumping the same also.

The great advantage gained by repeated compressions is the certainty of being able to prevent the maximum temperature ever exceeding that due to the maximum ratio of compression in any stage, by allowing it to cool

Diminution of temperature, principal object gained by compression at several stages.

down to its initial temperature after each stage of the compression. Air cannot in fact be compressed in one stage adiabatically to very high pressures, because the heat generated would interfere with the working of the compressor.

If the total number of thermal units abstracted during the actual process of pumping by several stages is equal to the number abstracted during pumping by a single stage, the work done in partial isothermal pumping will be the same in both methods, but if, owing to the longer period occupied in pumping by several stages, a greater number of thermal units is abstracted than in pumping by one stage, the work of partial isothermal pumping in the latter case will be greater than in the former.

Work due to machinery friction by several stages theoretically the same as by one stage.

As the total work done by the engine in both adiabatic and isothermal pumping is independent of the number of stages, the machinery friction ought theoretically to be independent of the number of stages, and in actual practice would be so to the extent to which the work due to friction depends upon piston pressure if the same sized compressors are used at each stage. The work due to friction in the motor-engine is however much less per horse-power in large than in small engines, so that the friction work overcome during several stages by separate engines would be much greater than by one stage. The best way to overcome this difficulty is to use a series of compressors arranged "tandem" fashion driven by the piston-rod of a single engine, the size of the cylinders being so arranged that the volume of the compressed air delivered by any compressor is equal to the contents of the cylinder of the succeeding compressor. The heat could easily be abstracted between each two stages by passing it through a coil of pipe surrounded by cold water.

Values of the product $p_0\ v_0$ and t_0 in the

It is usual to take p_0 equal to 14·7 lb. to the square inch, and to take a foot as the unit of measurement, so that case of free air generally adopted in numerical evalutions.

$$p_0 v_0 = 14 \cdot 7 \times 144 \, v_0 \text{ foot-lbs.}$$
$$= 2116 \cdot 8 \, v_0 \text{ foot-lbs.}$$

The temperature of the free air is usually taken at 60° F., or 521° F. on the absolute scale.

The specific gravity of the air at this temperature under a pressure of 14·7 lbs. to the square inch is ·0764. One cubic foot of air under 14·7 lbs. per square inch pressure and 60° F. temperature weighs therefore ·0764 lbs., and 1 lb. of air contains 13·09 cubic feet.

Direct application of heat to impart energy to air.

The energy imparted to air by the direct application of heat cannot in practice be utilised, except in conjunction with the work of an air-pump, unless it is alternately heated and cooled in the working cylinders of the motor. Heat alone therefore cannot be used as an agent for creating power to be transmitted to a distance. When used in conjunction with an air-compressor, the apparatus for heating the air ought to be placed close to the executive engine to prevent loss by conduction.

Work done in heating air in a closed vessel.

The number of thermal units required to raise the temperature of 1 lb. of air in a closed vessel is equal to the specific heat at constant volume, viz. ·1688, and the mechanical equivalent is therefore equal to ·1688 J.

Work done in heating air in a vessel which is capable of expansion when subjected to any assigned pressure.

When heat is applied to raise the temperature of the contained air subjected to the assigned pressure, the rise in temperature is accompanied by an increase in volume.

In expanding work is done equal to the increase in volume multiplied by the pressure and the number of thermal units required to do this work and raise the temperature of one pound of air one degree is equal to the specific heat of air at constant pressure, viz. ·2377, and the mechanical equivalent is therefore ·2377 J.

The number of thermal units whose mechanical equivalent is equal to the work done in adiabatic compression.

If after adiabatic compression the temperature of the air is reduced to its initial temperature the absolute energy of the air after cooling will be the same as before compression, no matter what the pressure after cooling may be. If reheated at constant volume to the temperature due to adiabatic compression, the whole of the energy imparted by adiabatic compression will be restored, and therefore the mechanical equivalent of the number of thermal units required to do this must be equal to the whole work done in adiabatic compression. The number of thermal units required to raise the temperature of the given weight of air $t_1 - t_0$ degrees at constant volume is equal to the weight in lbs. multiplied by $\cdot 1688\ (t_1 - t_0)$, and therefore, when the free air is subjected to a pressure of 14·7 lbs. to the square inch, and the initial temperature is 60° F., the work done in compressing any volume v_0 will be equal to

$$\begin{aligned}\cdot 0764\ v_0 \times \cdot 1688\ (t_1 - t_0)\ \mathrm{J} &= \cdot 0129\ v_0\ (t_1 - t_0)\ \mathrm{J} \\ &= \cdot 0129\ v_0\ (\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1)\ t_0\ \mathrm{J} \\ &= 6\cdot 721\ v_0\ (\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1)\ \mathrm{J}.\end{aligned}$$

Since the value 521 for t_0 corresponds with the indicated temperature of 60°.

A comparison of the values of the work done in adiabatic compression given by the formula in the terms of $p_0\ v_0$ with that given by the formula in terms of J, a means of comparing the accepted values of J and γ.

The approximate value of the mechanical equivalent of one thermal unit may be ascertained by comparing the value of the total work done in adiabatic compression given by the formula in terms of the pressure, volume, and γ with that given by the formula in terms of J and γ. Thus putting $v_0 = 1$, we have, since $p_0 = 2116\cdot 8$ lbs.,

$$\frac{2116\cdot 8\ (\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1)}{\gamma - 1} = 6\cdot 721\ (\mathrm{R}^{\frac{\gamma-1}{\gamma}} - 1)\ \mathrm{J},$$

whence

$$\mathrm{J} = \frac{314\cdot 95}{\gamma - 1}\ \text{foot-lbs.}$$

If $\gamma = 1 \cdot 408$, then $J = 771 \cdot 93$ foot-lbs., a remarkable coincidence of results of experiments totally distinct in their nature, the value of J being determined by Joule from observation of the effect produced by mechanical agitation on the temperature of water, and that of γ by the results of experimentally ascertaining the values of the specific heat of air. If $\gamma = 1 \cdot 401$, then $J = 785 \cdot 4$ foot-lbs.

Heat required to raise the temperature to the maximum temperature due to adiabatic compression, the same for all initial temperatures.

If w be the weight of the volume v_0 of compressed air at temperature t_0, its weight at any other temperature $t + t_0$ will be equal to $\frac{w\,t_0}{t + t_0}$, and the rise in temperature due to compression $\left(R^{\frac{\gamma-1}{\gamma}} - 1\right)(t + t_0)$, so that the number of thermal units required to raise the temperature will be equal to the product of the specific heat multipled by

$$\frac{w\,t_0}{t + t_0} \times \left(R^{\frac{\gamma-1}{\gamma}} - 1\right)(t + t_0) = w\,t_0\left(R^{\frac{\gamma-1}{\gamma}} - 1\right),$$

and is therefore the same for all values of the initial temperature.

All purely caloric air engines, apart from the losses due to conduction of heat to other bodies, necessarily uneconomical in working.

When energy is imparted to the air by the direct application of heat alone, the whole increase of energy is due to the number of thermal units actually taken up by the air; but when energy is imparted by mechanical compression, part of the increase of total energy is due to the work done by the pressure of the air on the back of the piston of the compressor. In the case, therefore, of expansive working after mechanical compression, the work done in overcoming atmospheric pressure is simply equal to the work done in compression by the same pressure, and therefore during perfectly adiabatic compression and expansion the whole of the work done by the engine is utilised; whilst in the case of energy imparted by heat, only the difference between the total energy imparted and the energy expended in overcoming the resistance of the atmosphere is utilised. If the air is heated in a vessel capable of expansion, the energy expended in increasing the volume is given back again if the heated air is utilised adiabatically. In estimating therefore the energy wasted by the use of heat, we must use the value of the specific heat at constant volume. This is necessarily wholly wasted in all caloric engines, in which the motive power is produced by the alternate heating and cooling of the air.

In the case of low pressures the work done by the air is nearly equal to the whole work done in compression. Even for a pressure of four atmospheres absolute, which is equal to a pressure of 44 lbs. per square inch indicated, the work done by the air is equal to the work done by the engine. See Column VI. Table I.

In the case of steam the resistance of the air can be almost entirely avoided by condensing the exhaust steam, and the heat abstracted partly utilised by feeding the boilers with condensation water. Against these savings have to be set the work done in pumping the condensing water. Although the loss of *absolute energy* by cooling after compression is equal to the whole increase of absolute energy imparted both by the external air pressure and the engine power, it is only the loss of *sensible* energy which affects the economics of the question. By reheating, although we expend, without reckoning loss due to conduction, energy equal to the whole absolute energy imparted during compression, we get back in useful effect only the equivalent of the work done by the engine in adiabatic compression, and lose the whole of the sensible energy possessed by the compressed air after cooling. There will also, as in the case of compression, be a still further loss due to the impossibility of using the heated air adiabatically *after* heating. However close the heating

When heat is used in conjunction with an air compressor for reheating after isothermal compression the loss is still greater.

apparatus may be to the motor, some of the heat imparted a second time must be lost by conduction.

Possible further loss suffered during reheating by irregular use of the compressed air.

The number of thermal units required to raise the temperature of the compressed air is the specific heat at constant pressure. The difference between this and the specific heat at constant volume causes the volume to increase, and the mechanical equivalent of the difference is therefore utilised when the air is actually being used, but in the case of a single engine during the periods of cut off, when the air is being worked expansively, the effect of reheating is to do work against the air in the supply pipe and force it back through the meter into the distributing mains. When, therefore, reheating is adopted a reflux valve ought to be used between the meter and the heating apparatus and an accumulator weighted to suit the maximum pressure provided.

Jacketing the only way to prevent loss by cooling.

Since the percentage of effective work got out of the energy actually imparted to the air by reheating is so insignificant (see Table VI.), it is clear that reheating, except by means of waste heat, ought never to be adopted. The only way in which the loss due to cooling can be diminished is to pump as nearly as possible adiabatically, and to prevent conduction of heat to the bodies surrounding the compressor and distributing pipes by coating these well with some non-conducting material.

Perfect adiabatic compression and expansion impossible.

However rapidly the process of air-compressing may be effected, the temperature of the cylinder and its surroundings must after a very short period of work be about the same as that due to compression, so that heat must be transmitted to the external air. Perfect adiabatic compression is therefore impossible. Similarly it is equally impossible practically to extract the whole heat generated during compression by using cooling water, because the period of compression is in the slowest engines too short to admit of this being effected. The nearest approach to isothermal compression is obtained by injecting sprays of water into the air cylinder. In Alpine regions, where there is an unlimited supply of water at a temperature very little above that of melting ice, a very near approach to isothermal compression may be effected by injecting sufficient ice-cold water into the cylinder of the compressor. In ordinary water-jacketed cylinders the water surrounding the cylinder always feels warm, i. e. its temperature is higher than 80° F. The external surface of the metal of the cylinder is necessarily hotter than the water, the internal than the external, and the compressed air than the internal surface.

Economy of the use of water for cooling doubtful unless it is supplied by gravitation free of cost.

Where a sufficient supply of cold water can be obtained without cost by gravitation, the utmost use ought to be made of this for securing the nearest possible approach to isothermal compression. The quantity of water required to insure the isothermal compression of a given weight of air depends upon the relative temperature of the water and the air before compression. The temperature of well-water will rarely be less than 60°, the ordinary initial temperature of the air. The author has not been able to find any record of the quantities of water used in practice, and therefore cannot form an estimate of the advantage to be gained by its use when it has to be pumped. Column VII. Table I. gives the ratios of the work done in perfect isothermal to that done in perfect adiabatic pumping, and Column II. Table II. the ratios of the estimated to the theoretical effective result, when allowance is made for initial heating.

The most perfect cooling during the act of pumping is effected by injecting the water into the cylinder, but this will add largely to the moisture contained by the air, and therefore to the snow deposited when the temperature during expansive working falls below the freezing-point. This increase in the quantity of snow will add greatly to the difficulty of utilising the sensible energy possessed by the compressed air. The diagrams taken during partial isothermal compression show that the increase of the temperature is ordinarily reduced to about one-half that due to perfect adiabatic compression, or the saving in the work done in the air cylinder effected by cooling for absolute pressures varying from $1\frac{1}{2}$ to 15 atmospheres, will vary from about 5 to 16 per cent. Water ought clearly never to be bought from a waterworks company for cooling purposes.

Sensible energy of the air after isothermal compression compared with the sensible energy after adiabatic compression.

Since the value of the product $p\,v$ is the same for all values of p so long as t remains constant, the value of this product after perfect isothermal compression cannot exceed $p_0\,v_0$; and the whole energy exerted during expansion from the pressure p_1 to p_0 will be equal to

$$\frac{p_0\,v_0\left(1 - \dfrac{1}{R^{\frac{\gamma-1}{\gamma}}}\right)}{\gamma - 1};$$

so that the sensible energy after adiabatic compression is equal to $R^{\frac{\gamma-1}{\gamma}}$ times the sensible energy after isothermal compression.

Maximum effective work attainable out of sensible energy.

The maximum effective work obtainable out of the sensible energy after perfect isothermal compression will be equal to the difference between the sensible energy and the work done in overcoming the resistance of the atmosphere. This last is equal to the pressure of the atmosphere multiplied by the volume swept out during expansion. This volume is equal to the difference between the volumes after and before expansion, viz. :—

$$v_0\left(\frac{1}{R^{\frac{\gamma-1}{\gamma}}} - \frac{1}{R}\right);$$

so that the maximum effective work realisable will be equal to

$$p_0 v_0 \left(\frac{1 - \dfrac{1}{R^{\frac{\gamma-1}{\gamma}}}}{\gamma - 1} - \frac{1}{R^{\frac{\gamma-1}{\gamma}}} + \frac{1}{R} \right)$$

$$= \frac{p_0 v_0}{\gamma - 1} \left(1 - \frac{\gamma}{R^{\frac{\gamma-1}{\gamma}}} + \frac{\gamma - 1}{R} \right). \qquad \text{IX.}$$

Maximum theoretical effective work which the air is capable of giving back after isothermal pumping.

The maximum theoretical effective work which the air is capable of giving back after perfect isothermal pumping is equal to the sum of the maximum effective work attainable out of the sensible energy, and the work done in forcing the compressed air into the receiver, viz.

$$\frac{(p_1 - p_0)\, p_0 v_0}{p_1}.$$

Since the whole work done by the engine in adiabatic pumping is equal to

$$\frac{\gamma\, p_0 v_0 \left(R^{\frac{\gamma-1}{\gamma}} - 1\right)}{\gamma - 1}$$

the ratio of the sensible energy to this is equal to

$$\frac{\left(1 - \dfrac{\gamma}{R^{\frac{\gamma-1}{\gamma}}} + \dfrac{\gamma - 1}{R} \right)}{\gamma \left(R^{\frac{\gamma-1}{\gamma}} - 1\right)}, \qquad \text{X.}$$

and of the work done in forcing the compressed air into the receiver to the same

$$\frac{(\gamma - 1)\left(1 - \dfrac{1}{R}\right)}{\gamma \left(R^{\frac{\gamma-1}{\gamma}} - 1\right)}; \qquad \text{XI.}$$

so that the ratio of the maximum effective work realisable after isothermal pumping to the work done by the engine in adiabatic pumping is equal to

$$\frac{1 - \dfrac{1}{R^{\frac{\gamma-1}{\gamma}}}}{R^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{R^{\frac{\gamma-1}{\gamma}}}. \qquad \text{XII.}$$

After proving that the sensible energy after isothermal compression is equal to $\dfrac{1}{R^{\frac{\gamma-1}{\gamma}}}$ times the sensible energy after adiabatic compression, this result is self-evident, because the isothermal volume of compressed air is equal to $\dfrac{1}{R^{\frac{\gamma-1}{\gamma}}}$ times its adiabatic volume.

Ratio of the minimum length of stroke required for the admission of the air to the full stroke.

If v be the volume of compressed air admitted into the cylinder of the executive engine, the expanded volume is equal to

$$\frac{R\,v}{R^{\frac{\gamma-1}{\gamma}}} = R^{\frac{1}{\gamma}}\,v,$$

and this must be equal to v_0, the volume of the cylinder, so that

$$\frac{v}{v_0} = \frac{1}{R^{\frac{1}{\gamma}}} \qquad \text{XIII.}$$

is the required ratio when the air is expanded to atmospheric pressure. When the pressure after expansion is greater than that of the atmosphere, the fall in temperature will be less, and if R_1 be the ratio of expansion, and the fall in temperature is equal to P per cent. of the fall corresponding to the ratio R, then R, R_1, and P will be connected by the following equation :—

$$\frac{P}{100}\left(1 - \frac{1}{R^{\frac{\gamma-1}{\gamma}}}\right) = 1 - \frac{1}{R_1^{\frac{\gamma-1}{\gamma}}},$$

whence

$$R_1 = \frac{R}{\left(\cdot oP + (1 - \cdot oP)R^{\frac{\gamma-1}{\gamma}}\right)^{\frac{\gamma}{\gamma-1}}} \qquad \text{XIV.}$$

and the required ratio of admission is

$$\frac{v}{v_0} = \frac{1}{R_1^{\frac{1}{\gamma}}}. \qquad \text{XV.}$$

Evaluation of the formulas.

We are now in a position to calculate the ratio of the theoretical maximum effective work realisable out of a given weight of compressed air to the actual work done in the cylinder of the air-compressor in adiabatic and isothermal pumping with absolute accuracy, the only unknown element being the exact value of γ. The difference between the extreme values of γ adopted by different scientific authorities is less than one-fortieth part of the maximum value. The divergence of the practical experimental values from the theoretical ones is due to the impossibility of insuring either perfect adiabatic or perfect isothermal pumping. The following is a description of the evaluations given in the columns of Table I.

I. The value of the ratio of compression.

II. The indicated pressure in lbs. per square inch.

III. The absolute temperature to which the air is raised by adiabatic compression, the initial temperature being 521° F.

IV. The absolute temperature to which the air falls after isothermal compression during expansion to the initial pressure, the initial absolute temperature being 521° F.

V. The ratio of the minimum length of the stroke required for the admission of the air to the full stroke for expansion to atmospheric pressure.

VI. The ratio of the work done by the engine in compression to the whole work done in adiabatic compression.

VII. The ratio of the work done in perfect isothermal to that done in adiabatic pumping.

VIII. The ratio of the work done by the engine in forcing the compressed air into the receiver during isothermal pumping to the whole work done by the engine in adiabatic pumping.

IX. The ratio of the maximum theoretical effective work realisable out of the sensible energy after isothermal compression to the whole work done by the engine in adiabatic pumping.

X. The sum of the ratios in columns VIII. and IX.

XI. The isothermal pumping ratio corresponding to the adiabatic pumping ratio in Column VIII.

XII. Ditto in Column IX.

XIII. Ditto in Column X.

TABLE I.

The values of the pressures and the temperatures are given to the nearest integer.

I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.	XIII.
1½	8	584	465	·71	·17	·98	·80	·09	·89	·81	·09	·91
2	15	641	424	·61	·31	·90	·63	·18	·81	·70	·20	·90
3	29	714	380	·46	·40	·86	·53	·21	·74	·62	·26	·88
4	44	782	347	·37	·49	·79	·43	·23	·66	·54	·29	·83
5	59	834	326	·31	·54	·77	·39	·23	·62	·50	·30	·80
6	74	886	307	·28	·58	·74	·35	·23	·58	·47	·31	·78
8	103	954	284	·23	·62	·73	·31	·23	·54	·42	·31	·73
10	132	1016	267	·20	·65	·70	·27	·23	·50	·38	·33	·71
15	206	1146	237	·15	·71	·66	·23	·23	·46	·33	·34	·67

Causes of divergences from the ratios in Table I. in actual practice.

The ratios in Table I. refer only to perfect adiabatic and perfect isothermal compression, which cannot be obtained in practice. Allowances have to be made for the effects of initial heating before compression and of imperfect isothermal pumping.

Initial heating of the air.

If we assume that the increase in the initial temperature of the air is equal to one-fourth of the increase due to adiabatic compression, the adiabatic ratios in Table I. must be multiplied by $\dfrac{4}{R^{\frac{\gamma-1}{\gamma}}+3}$.

Imperfect isothermal pumping.

From the actual diagrams taken during pumping, it appears that the maximum increase in temperature is reduced by about one-half. We may therefore look upon the actual working ratio as a mean between the adiabatic and isothermal ratios.

Percentage of sensible energy possessed by the air after isothermal compression realisable.

In order to admit of the utilisation of the sensible energy after isothermal compression means must be provided for dealing with the snow formed out of the moisture in the air. In the actual practice of refrigeration by means of the expansion of compressed air, it is found that with a pressure of about four atmospheres absolute the temperature of the air in the expansion cylinder rarely falls to $-55°$ below zero, although it falls by subsequent expansion to $-100°$ below zero in the freezing chamber. In the expansion cylinder the minimum temperature is therefore about 406°, or the fall from the initial temperature of 521° about 115°. The temperature after expansion down to atmospheric pressure (see Table I. Column IV.) is 347°, or the fall 174°. Since the energy given out varies with the fall in the temperature, the percentage of sensible energy, which can be utilised in practice, is equal to

$$\frac{115}{174} \times 100 = 66 \text{ per cent.}$$

for this degree of compression. If we adopt this percentage in all cases we must multiply the expansion ratios in Table I. by ·66. For this percentage of utilisation, R_1 the ratio of expansion is equal to

$$\frac{R}{\left(\cdot 66 + \cdot 34\, R^{\frac{\gamma - 1}{\gamma}}\right)^{\frac{\gamma}{\gamma - 1}}}.$$

Since the pressure on the back of the piston can be almost totally removed by condensing the exhaust steam, the steam may be expanded till its absolute pressure is fully 10 lbs. below that of the atmosphere. Owing also to the greater specific heat of steam there is a much less fall in the temperature during expansion than in the case of air. Thus the temperature of the air in expanding from a pressure of 44 lbs. to 14·7 lbs. falls from 521° to 347°, a fall of 174°; whilst in the case of steam, in expanding between the same limits of pressure the temperature falls from 274° to 212°, a fall of 62°.

Reasons why steam is more suited for expansive working than compressed air.

Table II.—The following is a description of the evaluations given in the columns of Table II.:—

I. The value of the ratio of compression.

II. Coefficients to allow for the effects of initial heating.

III. The ratio of the minimum length of stroke required for the admission of the air to the full stroke in expansive utilisation.

IV. The ratios in Column IX. Table I. multiplied by ·66, and the ratios in Column II. Table II.

V. The ratios in Column XII. Table I. multiplied by ·66.

VI. The ratios in Column VII. Table I. multiplied by the ratios in Column II. Table II.

VII. The ratio of the work done by the engine in adiabatic pumping to the work done in forcing the compressed air into the receiver after isothermal compression.

VIII. The mean of the ratios in Column VI. Table II. and Column XI. Table I.

IX. The ratio of the work done by the engine in partial isothermal pumping to the work done in forcing the compressed air into the receiver after isothermal compression.

X. The sum of the ratios in Columns IV. and VI. Table II.

XI. The ratio of the work done by the engine in adiabatic pumping to the whole effective work realisable.

XII. The sum of the ratios in Column XI. Table I. and Column V. Table II.

XIII. The mean between the ratios in Columns X. and XII. Table II.

XIV. The ratio of the work done by the engine in partial isothermal pumping to the whole effective work realisable.

TABLE II.

I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.	XIII.	XIV.
1½	·97	·84	·04	·06	·78	1·28	·80	1·25	·82	1·22	·88	·85	1·18
2	·95	·74	·11	·13	·60	1·67	·65	1·54	·71	1·40	·83	·77	1·30
3	·91	·61	·13	·17	·48	2·08	·55	1·82	·61	1·64	·79	·70	1·43
4	·89	·55	·14	·19	·38	2·63	·46	2·17	·52	1·92	·73	·62	1·61
5	·87	·51	·13	·20	·34	2·94	·42	2·37	·47	2·13	·70	·58	1·72
6	·85	·47	·13	·20	·30	3·33	·38	2·63	·43	2·32	·67	·55	1·82
8	·83	·42	·13	·20	·26	3·85	·33	3·03	·39	2·56	·62	·50	2·00
10	·80	·39	·12	·21	·22	4·55	·30	3·33	·34	3·00	·59	·46	2·18
15	·77	·34	·12	·21	·18	5·55	·25	4·00	·30	3·34	·54	·42	2·39

Type of engine required for expansive working after isothermal compression.

In order that the air may be used for expansive working, it will be necessary to provide means for dealing with the snow. The engines must also be capable of admitting the air during the whole length of the stroke. Ordinary stationary steam engines cannot as a rule admit the steam for more than one-third of the stroke; such engines cannot, therefore, be worked with air at a less pressure than 200 lbs. to the square inch (see Column III. Table II.) unless the expansion gear is altered.

Machinery friction and loss by slip.

The ratios in Tables I. and II. refer solely to the actual net work done in the air cylinder. The motive power necessary to produce these results is always greater than the power actually expended in the air-cylinder, because it has to overcome machinery friction. In the case of large compressors of the best description the work done by the engine does not exceed the work done in actual pumping of air by more than 20 per cent., but is much greater when the compressor is small. There is also in all cases a certain amount of slip past the valves before they close, and past the piston, so that taking both together 25 per cent. is the least that must be added to the whole work done in the air cylinder to get the minimum working power of the engine.

Margin of power required to cover possible mistakes in calculation of work and to insure economical working.

Since the power of the engine must be sufficient to do the work of air-pumping under all conditions, it is clear that the work done in adiabatic pumping must be made the basis of calculation, and the effective work must for the same reason be taken to be equal to the work done in forcing the air into the receiver after isothermal compression. In the great majority of cases in which compressed air is used expansive working is impossible, and in no case is it possible unless either the air is reheated or the usual means provided for dealing with the snow formed during the expansive working of air at a temperature below the freezing point.

In calculating the maximum engine power required we must use therefore the ratio of work done in the air cylinder to effective power produced given in Column VII. Table II. In designing pumping installations it is usual to provide engine power equal to double the maximum work to be done in the cylinder of the pump. If the increase in work due to machinery friction and loss by slip does not exceed 25 per cent., the power thus provided will have to the maximum work to be done the ratio

$\frac{2}{1 \cdot 25} = 1 \cdot 6$, or there will be a margin of 60 per cent., which is certainly not too much to cover risks of mistakes and insure economy of working. If the engines and boilers have to be pressed to anything like their maximum capacity, economical working is out of the question. If the increase in work due to friction and loss by slip amounts to 50 per cent., the ratio of maximum power to maximum work becomes $\frac{2}{1 \cdot 5} = 1 \cdot 34$, or there is only a margin of 34 per cent.

The indicated power of the engine is denoted by the letters I.H.P., and the effective result produced by H.P. The maximum value of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$, which gives the engine power to be provided, has in the following table been estimated to be double of the ratio in Column VII. Table II. This ratio may be called the safe ratio.

TABLE III.

	I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.
Value of R	$1\frac{1}{2}$	2	3	4	5	6	8	10	15
Safe value of $\frac{\text{I.H.P.}}{\text{H.P.}}$	2·56	3·34	4·16	5·26	5·88	6·66	7·70	9·10	11·10
Adiabatic value of $\frac{\text{I.H.P.}}{\text{H.P.}}$ without expansion ..	1·60	2·09	2·60	3·30	3·68	4·16	4·81	5·69	6·94
Minimum working value of $\frac{\text{I.H.P.}}{\text{H.P.}}$ without expansion	1·56	1·93	2·28	2·72	2·97	3·30	3·80	4·17	5·00
Minimum working value of $\frac{\text{I.H.P.}}{\text{H.P.}}$ with expansion	1·48	1·63	1·79	2·00	2·20	2·28	2·50	2·73	3·00

Value of the working ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$.

Since in actual practice of air-compressing it is found that the increase in temperature is reduced by about one-half in partial isothermal compression, the value of the working ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$ will probably be about a mean between the adiabatic and isothermal ratios, or will be equal to the ratios in Columns XI. and XIV. Table II. increased by the percentage due to machinery friction and loss by slip. If therefore we add 25 per cent. to these ratios we shall get the minimum possible working ratios of power to effective work when the air is used with or without expansion. They are the ratios given in Table III.

Power required to pump 100 cubic feet of free air.

In several of the cases, where compressed air is used, it is impossible to calculate the value of the effective work which a given amount of compressed air can do, such as the work done in driving boring machinery, but it is known what volume of compressed air is required. The following table gives the safe and working I.H.P. required to pump 100 cubic feet per minute of free air compressed to pressures varying from 1½ to 15 atmospheres absolute. For ordinary work air is rarely compressed to more than 5 atmospheres absolute, so that the table covers all the cases likely to occur. For pressures of less than 1½ atmospheres absolute the work done in the air cylinder in pumping will be practically equal to the contents of the cylinder multiplied by the indicated pressure. When the air is used without expansion, as it nearly always necessarily must be, the *effective* work accomplished is equal to the work of forcing the compressed air into the receiver after having been cooled, since cooling to the temperature of its surroundings must necessarily take place before it is used. The effective work accomplished is therefore constant for all pressures, and is equal to 6·4 H.P. The horse-powers are given to the nearest H.P. in Table IV.

Percentage of work realisable out

The percentage of work realisable out of the whole potential energy imparted to the compressed air which of the whole potential energy imparted to compressed air.

remains in it after being cooled down to the initial temperature will vary with the distance to which it has to be transmitted and the nature of the work to be done.

TABLE IV.

Value of R	1½	2	3	4	5	6	8	10	5
Effective H.P produced ..	6½	6½	6½	6½	6½	6½	6½	6½	6½
I.H.P. partial isothermal pumping	10	12	14	17	19	21	24	27	32
I.H.P. adiabatic pumping	10	13	17	21	24	27	31	36	44
I.H.P. safe	16	21	26	34	38	42	50	58	71

Without knowing the exact circumstances of each case it is impossible to form an estimate of the ultimate efficiency. Taking account of executive engine friction, loss by friction in the pipes, and leakage, it will not be safe to calculate on realising more than 60 per cent. of the potential energy possessed by the air after cooling. We shall get therefore probably minimum values of the safe and working values of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$ when the H.P. represents the final effective work done by the executive engine, by dividing the values in Table III. by ·6. In the following table the ratios are given to the nearest quarter of a unit.

TABLE V.

Value of R	1½	2	3	4	5	6	8	10	15
Safe value of $\frac{\text{I.H.P}}{\text{H.P.}}$. ..	4¼	5½	6¾	8¾	9¾	11	13	15	19
Adiabatic value of $\frac{\text{I H.P.}}{\text{H.P.}}$ without expansion ..	2¾	3½	4¼	5½	6	7	8	9½	11½
Minimum working value of $\frac{\text{I.H.P}}{\text{H.P.}}$ without expansion	2½	3¼	3¾	4½	5	5½	6¼	7	8¼
Minimum working value of $\frac{\text{I.H.P.}}{\text{H.P.}}$ with expansion	2½	2¾	3	3¼	3¾	4	4¼	4½	5

The question of heat has already been so fully discussed that little more need be said, but we are now in a position to give in exact figures the maximum percentage of work which can be got out of the absolute energy imparted to air by the direct application of heat, leaving out of consideration all losses due to the conduction of heat.

Maximum theoretical percentage of the total energy imparted to air by the direct application of heat realisable.

The absolute energy of a given weight of air varies as its absolute temperature, and is therefore the same whether the rise in temperature is produced by compression or by the direct application of heat; but in the latter case the whole increase is due to the applied force, viz. heat, whilst in the former it is only the increase in sensible energy which is produced by the applied power, and it is the sensible energy only which can be utilised. The ratios therefore in Column VI. Table I. multiplied by 100 give the maximum percentage realisable out of the *actual energy imparted to the air* at constant volume by heat, if there is no *subsequent* loss by conduction.

After compression, the air, when cooled down to its original temperature, still possesses sensible energy, and it is only the difference between the sensible energy possessed by the air before and after reheating which would be imparted to the air by reheating. There is no evidence to show that a greater percentage of the expansive energy possessed after reheating can be obtained in practice than out of the sensible energy possessed by the cold air, if suitable means are provided for dealing with the snow.

Evaluation of the maximum net percentage realisable out of the absolute energy imparted to the air by reheating after isothermal pumping.

The ratios in Column IX. Table I. multiplied by 100 give the percentages of the effective work realisable out of the sensible energy possessed by the air after cooling in terms of the work done by the engine in pumping, and as this is γ times the whole work done in adiabatic compression, these percentages multiplied by γ will give the percentage of the effective work realisable out of the sensible energy possessed by the air after cooling in terms of the whole work done in adiabatic compression, i. e. in terms also of the number of thermal units required to reheat the air at constant volume. The following table gives the percentages of adiabatic and isothermal sensible energy, and the difference between them, which represents the maximum net percentage which can be got out of the absolute energy imparted by reheating when there is no loss by conduction of heat to other bodies:—

TABLE VI.

Value of R..	1½	2	3	4	5	6	8	10	15
Adiabatic sensible energy	17	31	40	49	54	58	62	65	71
Isothermal sensible energy	13	25	30	32	32	32	32	32	32
Percentage of absolute energy imparted by heat realisable	4	6	10	17	22	26	30	33	39

Efficiency of heating apparatus necessarily small.

The heat must necessarily be conveyed to the air by the medium of the metal passages through which the air flows. Gas cannot be burned in the interior of an air receiver unless it is kept at a pressure slightly higher than that of the compressed air, or in other words it must be especially pumped. In heating, therefore, the losses by conduction must be very great and the efficiency of the heating apparatus very small.

Loss of heat by conduction occurs at two periods.

When the air is heated by compression, the thermal units lost by cooling are simply the equivalent of the whole work done in compression, that is, to the absolute energy imparted to the air, but by the process of heating it is only a part of the thermal units expended which are conveyed to the air itself. In the case of compression, the loss due to machinery friction may be set against the loss due to conduction in the direct application of heat. The value of the former loss can always be ascertained, that of the latter never.

Power cannot be stored up by means of compressed air.

Unless the compressed air is used as fast as it is produced, it is only the energy given out by the air, whilst the pressure in the receiver falls from the maximum to the minimum required to do the work, which represents the available stored-up energy. Its amount is necessarily very small. In addition to the increase in work entailed by compressing the air to a higher tension than is necessary, there will be a further loss due to cooling during expansion to the lower pressure before it reaches the executive engine. Accumulators cannot be used for storing up air at ordinary working pressures, because the volume required is too large.

Variation of work also entails heavy waste of energy.

When air of a minimum pressure is required to do a varying quantity of work, it is necessary to keep it at a pressure much higher than the minimum required to do the work when the quantity of work is least, in order that the volume required when the quantity of work is greatest may be supplied by expansion. There is, therefore, a loss similar to that sustained when power is attempted to be stored up for future use. When the variation in the rate of work is small, this loss may be obviated by the use of accumulators.

CHAPTER II.

EFFICIENCY OF HIGH PRESSURE WATER.

Fluids under pressure transmit power simply as fluid pistons.

WHEN an elastic fluid is used for the transmission of power, no power can be transmitted until the minimum pressure required to do the work has been attained by actual compression. The investigations in Chapter I. have shown how costly is the work of first forming this fluid piston. When an incompressible fluid like water is used, this piston is already formed, and the whole of the engine power expended actually transmits power.

Machinery friction to be overcome in pumping elastic compared with that to be overcome in pumping incompressible fluids.

The only difference which can arise in the work due to machinery friction when expressed as a percentage of the *whole work* done in the pump barrel, must be caused by variation in piston friction work. If the work due to piston friction bears in both cases the same ratio to the total pressure on the piston, the whole work due to machinery friction will be the same in both cases, when the *whole work* done in the pump barrel is the same. In estimating, however, the value of the work due to friction in terms of the *net effective result* produced, measured by the *product of the volume multiplied by the pressure* in each case, the percentage of work due to friction in the case of air will be equal to the percentage in the case of water multiplied by the ratios in columns VII. and IX. Table II. the adiabatic and partial isothermal ratios respectively. Before therefore we can compare the total work done in the two cases, it is necessary to investigate the question of piston friction.

Piston friction varies with the area of the rubbing surfaces in contact multiplied by the pressure per unit of area. In similarly constructed pistons therefore the frictional resistance must vary as the diameters. When leather packing is used, the unit pressure in question is equal to the pressure of the fluid in the pump barrel, and in all cases it is proportional to it, so that, if F represents the resistance due to piston friction, p the pressure per unit of area in the pump, and d the diameter, we shall have

Percentage of piston friction in terms of the total pressure on the piston independent of the pressure per unit of area.

$$F = f p d,$$

where f is a constant, the value of which varies with the nature of the packing. Since the total pressure on the piston is equal to $\cdot 78\ p d^2$, the ratio of F to the total pressure is equal to

$$\frac{f p d}{\cdot 78\ p d^2} = \frac{c}{d},$$

where c is a constant. The percentage, therefore, of the friction in terms of the total pressure on the piston is independent of the magnitude of the pressure per unit of area of the piston.

In cylinders of different capacities, in which the ratio of the length of stroke to the diameter is constant, the diameters vary directly as the cube roots of the contents of the cylinders. If therefore F_1 F_2 represent the piston friction work overcome in a single stroke, corresponding to the cylinder contents v_1, v_2 respectively, we shall have

Comparison of piston friction work in terms of the volumes pumped.

$$\frac{F_1}{F_2} = \frac{\sqrt[3]{v_2}}{\sqrt[3]{v_1}}.$$

When hemp or similar packing tightened by a gland is used, the amount of the frictional resistance varies with the tightness with which the gland is screwed down.

Value of piston packing friction.

When metallic or leather packing is used, the frictional resistance is independent of the tightness of the gland screws, and therefore the results of experiments to ascertain the amount of this friction will be applicable to all cases. Since the friction between rubbing surfaces is greater when they are dry, than when they are wet, the friction per unit of area of piston packing, will *cæteris paribus* be greater in the cylinder of an air compressor, than in that of a water pump. Thus the resistance of leather on gun-metal is equal to 56 per cent. of the pressure when dry, and only to 36 per cent. when wet. Leathers ought not to be deeper than is necessary to secure the tightness of the packing. With very high pressures, therefore, a less depth in contact with the surface, which slides over the leather, will suffice than with low pressures. In any case half an inch in contact subjected to the pressure of the liquid will suffice, and the frictional resistance will be given by the following equation

$$F = \frac{f \pi d p}{2},$$

where f is the coefficient of friction. In the case of wet leather $f = \cdot 36$ and

$$F = \cdot 57 \, p \, d.$$

According to the result of experiments made by Messrs. Hicks of Bolton,* the value of F is given by the equation

$$F = \cdot 471 \, p \, d,$$

when the leathers are new, and by the equation

$$F = \cdot 313 \, p \, d,$$

when the leathers have been in use. The equivalent depth of the leathers in contact under full pressure must there-

* See Spons' 'Dictionary of Engineering,' article 'Hydraulic Machines.'

fore have been equal to $\frac{\cdot 47}{\cdot 57} \times \cdot 5 = \cdot 4$ in. They found also that the frictional resistance did not vary with the depth of the leathers—a result which can only be explained in this way, viz. that it is only the lips of the leathers, which actually press against the metal sliding over them, the rest being kept from contact by the pressure of the glands. If we adopt the values ascertained by Messrs. Hicks as reliable, the value of F expressed as a percentage of the total pressure, since the total pressure is equal to $\cdot 78\, p\, d^2$, will be

$$F = \frac{60}{d} \text{ for new leathers,}$$

$$F = \frac{40}{d} \text{ for used leathers.}$$

In the case of small pistons a much less total depth of leather is necessary than in the case of large pistons to insure efficient working. With very high pressures and small pistons, a contact of one-quarter of an inch will suffice, so that the corresponding values will be about one-half those given by Messrs. Hicks.

Total work, including machinery friction, to be overcome in pumping elastic compared with that to be overcome in pumping incompressible fluids when the effective result expressed in terms of the product of the pressure multiplied by

In order to simplify the investigations, we will assume that the ratio of the length of the stroke to the diameter is the same both in the air and water pump. Whatever ratio is best in one case is best in the other, and therefore the assumption is a legitimate basis for calculation. Let the symbols explained in Chapter II. have the same meaning, and let

D, d, be the diameters in inches of the water and air pump respectively.

m the ration of the length of stroke to the diameter.

P the maximum absolute pressure in the water pump per square inch.

the volume produced in a given time is the same.

V the volume of the water pumped.

R_1 the ratio $P \div p_1$ of the maximum absolute pressure of the water to the maximum absolute pressure of the air.

r the ratio of the number of strokes made by the air pump to those made by the water pump per minute.

Since v_1 is the volume of the compressed air after isothermal compression, the condition of equality of effective work is

$$P V = p_1 v_1 = p_0 v_0,$$

and we also have

$$\frac{\pi m D^3}{4} = V = \frac{v_1}{R_1} = \frac{\pi r m d^3}{4 R_1 . R},$$

whence

$$D = \frac{d \sqrt[3]{r}}{\sqrt[3]{R . R_1}}.$$

Now the work done by the engine bears in both cases the same ratio to the *whole work* done in the pump cylinders. In the case of the water pump, this whole work is equal to the sum of the effective work, and the work due to friction or to

$$P V \left(1 + \frac{C}{D}\right) = p_1 v_1 \left(1 + \frac{C \sqrt[3]{R . R_1}}{d \sqrt[3]{r}}\right).$$

In the case of the air engine, the whole work is equal to the sum of the effective work, and the friction work corresponding with the effective work multiplied by the ratios in Columns VII. and IX. of Table II. for adiabatic and partial isothermal work respectively. Denoting these coefficients by the symbol C_1, the whole work done by the engine in air pumping will be equal to

$$C_1 p_1 v_1 \left(1 + \frac{C}{d}\right),$$

if the piston friction constant, C, is the same in both cases. Therefore the ratio of the whole work done by the air-pump engine to the whole work done by the water-pump engine will be equal to

$$\frac{C_1\left(1 + \frac{C}{d}\right)}{1 + \frac{C\sqrt[3]{R.R_1}}{d\sqrt[3]{r}}}.$$

As air-pump engines usually make about four times as many strokes as water-pump engines in the same interval of time, we may adopt the value 4 for r. We may also regard the value 60 for C as one that will not be exceeded for *wet* leather packing in the case of the water pump, and we shall not prejudice the case of air by adopting this value. With these values of C and r, the ratio is equal to

$$\frac{C_1(d + \cdot 6)}{d + \cdot 377\sqrt[3]{R.R_1}}.$$

Since $R = \frac{p_1}{p_0}$ and $R_1 = \frac{P}{p_1}$ the product $R.R_1 = \frac{P}{p_0}$, and is therefore independent of the value of p_1, the absolute pressure of the compressed air, and depends solely on the absolute pressure of the water. When $R_1 = 1$, or the absolute pressure of the water is the same as that of the compressed air, the value of the ratio varies with the value of R. The following table gives the values of the ratio

$$\frac{d + \cdot 6}{d + \cdot 377\sqrt[3]{R_1 R_1}}. \qquad \text{A.}$$

for values of R R_1, varying from $1\frac{1}{2}$ to 100 for 9 and 24 inch pipes, the ratios being either that of the absolute pressure of the water to that of the free air; or, when the absolute pressure of the water is equal to that of the compressed air, the ratios of compression.

TABLE VII.

Value of $R R_1$	$1\frac{1}{2}$	4	15	30	60	100
Value of P in lbs. per square inch	22	59	221	541	882	1471
Value of A for 9-inch air pump	1·18	1·00	·97	·95	·92	·89
Value of A for 24-inch air pump	1·01	1·00	·99	·98	·97	·96

The coefficients given in the above table are those by which we must multiply the ratios given in Columns VIII. and IX. Table II. to find the value of the ratio.

$$\frac{\text{Work done in air pumping}}{\text{Work done in water pumping}} \qquad \text{E.}$$

when the same effective result, leaving the sensible energy of the compressed air out of consideration, is achieved in both cases. The value of these coefficients is so nearly equal to unity, that for all practical purposes the ratios in Table II. give practically the values of the ratio E. A reference to the formula and the evaluations shows that for all degrees of compression up to that which is equal to the ratio of the speed of the air pump to that of the water pump, the ratio E exceeds the corresponding ratio in Table II., so that the greater the speed of the air compressor the less the efficiency, so far as this is affected by machinery friction.

Safe and working values of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$ in the case of the pumping engine..

In the case of water the effective work done in the pump-cylinder, leaving friction of machinery out of consideration, is equal to the whole work done. If, therefore, we adopt the same rule for determining the safe and minimum I.H.P. in the case of water, as in the case of air, which the preceding investigations show that we ought to do, we get—

$$\text{Safe value of ratio } \frac{\text{I.H.P.}}{\text{H.P.}} = 2:1,$$

$$\text{Minimum working ditto, } = 1\tfrac{1}{4}:1.$$

Ultimate safe and working values of ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$.

As in the case of air it is impossible to form any estimate of this without knowing the kind of work which the high-pressure water has to perform. It will be proved in the next chapter that the percentage of loss due to pipe friction in conveying the same amount of power is less than the same percentage in the case of air, so that if we adopt for the purposes of comparison the same value of the effective work obtained, viz. 60 per cent. of the power transmitted, we shall not err in favour of high-pressure water. With 60 per cent. efficiency we get—

$$\text{Safe value of ratio } \frac{\text{I.H.P.}}{\text{P.}} = 3\tfrac{1}{4}:1,$$

$$\text{Minimum ditto} = 2:1$$

for all pressures.

Accumulators for storage of power and equalising the work of the engine available.

The use of high-pressure water admits of the employment of accumulators, because a small volume of water represents a great deal of power—leaving the sensible energy of the air out of consideration, 100 gallons of water at from 800 to 900 lbs. pressure represent as much effective work as 2000 gallons of air at from 40 to 50 lbs. pressure.

With accumulators the pressure remains constant, and, therefore, no work is wasted as in the case of air, which can only be used as a reserve power by storing it up in receivers, compressed to a much higher pressure than is necessary to do the work.

CHAPTER III.

RELATIVE ADVANTAGES OF AIR AND WATER FOR TRANSMISSION OF POWER.

Pipe friction causes a loss of absolute as well as of sensible energy in compressed air.

WHEN an elastic fluid is forced through a pipe, the frictional resistance alone causes a loss of pressure, and the work done by the air in overcoming frictional resistance is equal to the mechanical equivalent of the thermal units lost by the fall in temperature, which is equal to $\left(1 - \left(\frac{p}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right) t_0$, p_1 being the absolute pressure, and t_0 the absolute temperature before the air enters the pipe, and p the pressure at the point under consideration. The absolute energy after overcoming the work due to friction will be to the initial absolute energy in the ratio of $1 - \left(\frac{p}{p_1}\right)^{\frac{\gamma-1}{\gamma}} : 1$ if no heat is restored.

Temperature partly restored through the heat generated by friction.

In accordance with the maxim that force is indestructible, the energy lost by the air must have been given back to itself, transferred to some other body, or partly given back to itself, partly transferred to some other body. It would in this instance be partly restored to the air by diminishing the loss of temperature due to expansive working, and partly expended in raising the temperature of the pipes.

Original temperature may be restored by conduction from pipes,

If t_0 is the same as the temperature of the pipes and the body surrounding them, the heat lost by the air after expansion will be again restored to it, and the absolute energy will remain unchanged, but the sensible energy will be to the initial sensible energy in the ratio of

but loss of sensible energy unavoidable.

$$\frac{1 - \left(\frac{p_0}{p}\right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}}.$$

When the air is used without expansion, the product $p\,v$ will be equal to the product $p_1\,v_1$, when the temperature is the same in both cases, but the work expended in the first instance in raising the air to the higher temperature will have been expended in overcoming pipe friction.

Loss of head due to pipe friction in the case of air.

Some authorities maintain that the frictional resistance due to the flow of gases in pipes is independent of the density; but this opinion is clearly untenable, since not only must the skin friction of the air against the pipes be increased by the pressure, but also the friction between the molecules of the air. For very low pressures we may possibly, however, consider the frictional resistance to be practically independent of the density. The following formula, modified from one given in Box's 'Treatise on Heat,' based on the assumption that friction is independent of the density, is easily evaluated. It is

$$H = \frac{\cdot 00016\, l\, v^2}{d};$$

where

H = head lost in inches of water;
v = velocity in feet per second;
l = length of pipe in yards;
d = diameter in inches.

Conversely, if H represent the maximum available head for overcoming the friction due to a length of 1000 lineal yards, then

$$v = 2\cdot 5\sqrt{d\,H}.$$

In paper No. 2345, Vol. XCIII., of the 'Minutes of Proceedings of the Institute of Civil Engineers,' Professor Unwin gives a formula for the velocity of flow in which the effect of the variation of pressure is taken into account. The formula is

$$v = c\sqrt{\frac{d}{l}};$$

where

v = velocity in feet per second;
d = diameter in feet;
l = length in feet.

The coefficient c has different values, which depend only upon the ratio of the initial to the final pressures, and not upon the difference of those values. The following values are given for c, corresponding to different values of the ratio $\frac{p}{p_1}$:—

$\frac{p}{p_1}$.	c.
·95	2085
·90	2912
·85	3518
·80	4007
·75	4418

If the velocity of the flow were independent of the density of the air, the rate of flow would vary as the square root of the difference between the initial and final processes. It follows, therefore, that, if Professor Unwin is right in maintaining that the rate of flow is independent of the difference between the initial and final pressures, the resistance due to friction must also vary as the square root of the same difference, or as the square root of the density.

For the sake of comparison, we may take $d = 1$ foot, $p_1 = 58 \cdot 8$ lbs. the equivalent of four atmospheres absolute, and $l = 8100$ lineal feet. According to Professor Unwin's formula, the velocity per second for a fall of 5 per cent. in the pressure would be for all the values of p_1 equal to $23 \cdot 2$ lineal feet. Since the absolute fall in pressure is equal to $2 \cdot 94$ lbs., the corresponding value of H is $81 \cdot 6$ inches, and the velocity, according to Box's formula, would be $47 \cdot 6$ feet per second. The value of p_1, which gives the same value in both formulæ, is $14 \cdot 7$ lbs., or is equal to the atmospheric pressure. For a pressure of four atmospheres absolute, the velocity, according to the formula of Box, is rather more than double the velocity according to the formula of Professor Unwin.

Loss of head due to pipe-friction in the case of water.

Since water is incompressible, the density is constant for all pressures, and the increase of frictional resistance due to increase of pressure must be caused solely by the "skin" friction of the external film of water against the sides of the pipes, and will therefore affect the relative frictional values more in small than in large pipes. The author does not know of any experiments which have been made to decide the question of possible increase of friction due to increase of pressure, but in treating the value of the frictional resistance in both cases as independent of the pressure, we shall most certainly bias the computation in favour of air. Kutter's formulæ for the flow of water are generally accepted as the most reliable, and are equivalent to the following, in which v is the velocity in inches per second and s the hydraulic inclination.

TABLE A.—FOR SMOOTH PIPES.

Pipes ½ in. to 2½ in. diameter	$v = 107\, d^{\cdot 9} \sqrt{s}$
„ 2½ „ 5 „	$v = 115\, d^{\cdot 8} \sqrt{s}$
„ 5 „ 10 „	$v = 134\, d^{\cdot 7} \sqrt{s}$
„ 10 „ 72 „	$v = 166\, d^{\cdot 6} \sqrt{s}$
„ 6 ft. to 400 ft. „	$v = 256 \sqrt{d s}$

TABLE B.—FOR MODERATELY SMOOTH PIPES.

Pipes ½ in.	to 2½ in.	diameter	..	..	$v = 63\,d\sqrt{s}$
„ 2½	„ 5	„	..	..	$v = 68\,d^{\cdot 9}\sqrt{s}$
„ 5	„ 10	„	..	..	$v = 78\,d^{\cdot 8}\sqrt{s}$
„ 10	„ 24	„	..	..	$v = 100\,d^{\cdot 7}\sqrt{s}$
„ 24	„ 96	„	..	..	$v = 138\,d^{\cdot 6}\sqrt{s}$
„ 8 ft.	to 400 ft.	„	..	..	$v = 221\sqrt{d\,s}$;

For the purpose of comparing the transmitting powers of air and water, the author has used the mean of the values given by Tables A and B. The larger values given by Table A for pipes of less than six inches diameter correspond very nearly with those given by Neville's formulas, which are based on the results of experiments with pipes of small diameter, and are, therefore, probably more nearly correct than the lesser values corresponding to the mean of the values given by Tables A and B. Thus in the case of small pipes ample allowance will have been made for any possible increase of frictional resistance due to increase of pressure.

Bursting effect due to sudden changes of momentum of a heavy incompressible fluid like water.

The bursting effect produced by a sudden stoppage of flow of the water varies with the sudden increase of pressure per unit of area caused by the sudden destruction of the momentum. This increase of pressure therefore is equal to the product of the sudden change in the velocity of flow multiplied by the mass of a column of water whose area is equal to the unit of area and length equal to that of the pipe. If we adopt a square inch as the unit of area, a column 144 feet long will contain one cubic foot of water and will weigh 62·5 lbs., so that the weight of any other length of column on lbs. will be equal to $\frac{62 \cdot 5\, l}{144} = \cdot 434\, l$ where l is the length in feet, and the sudden increase on the pressure per square inch due to a sudden change in the velocity per second will be

$$\frac{\cdot 434\, l\,(v_1 - v_2)}{g} = \cdot 0132\, l\,(v_1 - v_2)$$

where v_1 is the velocity in feet per second before, and v_2 after the sudden shock, and g the accelerating force of gravity per second.

Since the extension or compression of an elastic material produced by a suddenly applied stress is equal to double the extension or compression caused by a gradually applied stress of the same intensity, the sudden increase of pressure must not exceed one-half of the difference between the proof and working stresses. The metal of the pipes ought always to be able to stand without risk of injury to elasticity a stress equal to double the working stress, so that, if P represents the working pressure in the pipes, we must have

$$\cdot 0132\, l\,(v_1 - v_2) \text{ not greater than } \frac{P}{2}.$$

Sudden stoppage to velocity of flow impossible in practice.

When there are several executive engines utilising the power, it is clear that the velocity of flow in the pipes cannot vary suddenly, since it is impossible that more than a very small percentage will stop and start simultaneously. In the case of one executive engine, the water will be flowing when the executive engine is at work, and stopping when it is at rest, but the change can in neither case be effected suddenly, unless the valve of the engine closes instantaneously. With slide valves this sudden changing is impossible. At the commencement of the stroke the gradual opening of the slide valve causes the velocity of flow to increase gradually, and its gradual closing stops its flow gradually. In any case, all risk of rupture due to sudden changes of velocity, where one executive engine only is used, can be obviated by using a well-charged air-vessel close to the executive engine. In designing installations for transmitting power by means of high-pressure water, therefore, the question of the bursting effect of sudden stoppages of velocity need not be taken into consideration, because they cannot occur.

High pressure does not impose any limit to the size of the pipes within the range of possible requirements.

If we were restricted to the use of cast iron in pipe-making, although there would be no difficulty in casting them of the requisite strength, the thickness of metal necessary would make them too heavy for use in laying mains, since the cost of transporting and laying them would be very great.

Bessemer steel is, however, now so cheap that it ought unquestionably to be used for all high-pressure mains larger than six inches in diameter, if not for the smaller sizes. If made of steel the metal of a pipe 24 in. diameter need not be more than one inch thick to stand a pressure of 800 lbs., since, when subjected to the proof test of 1600 lbs. per square inch, the tensile stress on the metal would be less than 9 tons to the square inch. To insure a minimum of one inch, an average thickness of $1\frac{1}{8}$ in. would be sufficient.

Proper mode of estimating the cost of the pipes.

In comparing the costs of the pipes in the case of air and water, the actual cost of the piping per lineal yard ought not to be taken as the standard, but the cost per H.P. conveyed by the pipes.

Comparison of the transmitting power of air and water pipes.

The calculations in the following table are based on an indicated working pressure of from 800 to 900 lbs. per square inch in the case of high pressure water, and of from 40 to 50 lbs. for the air, the condition being that the loss of pressure due to friction is not to exceed 5 per cent. of the initial indicated pressure in a distance of 6000 lineal yards. In the case of water this would be equivalent to a hydraulic inclination of about 1 in 180, and the value of H would be the number of inches of water corresponding to a pressure of ·37 lbs., or about 10·3 inches. The columns in Table VII. contain the following particulars:—

I. Diameter of pipes.

II. Velocity in feet per second of the water.

III. H.P. transmitted by water per minute equal to volume × 800 lbs. ÷ 33,000 lbs.

IV. Velocity in feet per second of the air.

V. H.P. transmitted by the air without expansive work equal to volume × 44 lbs. ÷ 33,000.

VI. H.P. transmitted by the air equal to the sum of H.P. in Column V. and 66 per cent. of the *effective* sensible energy after cooling.

VII. Ratio of H.P. in Column III. to H.P. in Column V.

VIII. Ratio of H.P. in Column III. to H.P. in Column VI.

IX. Size of pipes for conveying by high pressure water the H.P. in Column V.

X. Ditto in Column VI.

XI. Ditto half the H.P. in Column VI.

XII. Maximum length in yards of pipe admissible without risk of damage by sudden stoppage of flow for working pressure of 800 lbs. per square inch.

XIII. Ditto of 100 lbs. per square inch.

The ratios in Columns VII. and VIII. give the ratio of the maximum price per yard of water pipes to air pipes, consistent with equality of cost per H.P. conveyed.

The addition in Column VI. to the H.P. in Column V. due to realisable sensible energy, is equal to the H.P. in Column V., multiplied by the coefficient in Column IV. Table II. and divided by the coefficient in Column VIII. Table I. For absolute pressures of 4 atmospheres equal to 44 lbs. per square inch indicated, the multiplier is equal to ·33.

Let Q_1 Q_2 be the volumes conveyed by pipes of diameters D_1 D_2, respectively, then must

$$\frac{D_1}{D_2} = \sqrt[5]{\frac{Q_1^2}{Q_2^2}}$$

if the hydraulic inclination is the same in both cases.

TABLE VII.

I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	IX.	XII.	XIII.
3	1·5	16	14	8	11	2	1·5	2¼	2½	2	6700	840
4	1·8	33	16	16	21	2	1·6	3	3¼	2½	5600	700
5	2·2	63	18	28	37	2·3	1·7	3½	4	3	4600	570
6	2·5	103	20	45	60	2·3	1·7	4¼	4¾	3¾	4000	500
8	3·1	227	23	94	125	2·4	1·8	5¾	6½	4¾	3200	400
10	3·6	408	26	162	216	2·5	1·8	7	8	6	2800	350
12	4·0	658	28	254	339	2·6	1·9	8¼	9¼	7	2500	300
18	5·2	1834	34	692	923	2·7	2·0	12	13¼	10½	1940	240
24	6·4	4194	40	1448	1931	2·9	2·2	16	17½	13	1570	200

A gradient of 1 in 180 corresponds with a fall of 42 lbs. in 6000 yards, or the initial indicated water pressure is 840 lbs. Similarly a fall of ·37 lbs. in 1000 yards corresponds with 2·22 lbs. in 6000 yards, or the corresponding initial indicated air pressure is 46·2 lbs. *For this pressure the velocities given by Professor Unwin's formula are only equal to one-half those given by Box's formula,* and the diameters given in Column XI. are those of pipes, which will transmit the same horse-power with water at 840 lbs. initial indicated pressure, that can be transmitted by air at 46 lbs. initial indicated pressure in pipes of the diameters stated in Column I., the horse-power in the case of air being *one-half of the maximum* stated in Column VI., in which expansive working is allowed for, if Professor Unwin's formula is correct.

In Table VII. the H.P. which can be transmitted by water at 840 lbs. initial indicated pressure is compared with the H.P. which can be transmitted by air at 46 lbs.

initial indicated pressure in pipes of the same diameter, when the loss of pressure has the same percentage of the maximum pressure in both cases. In order to complete the comparison it is necessary to calculate the loss of pressure entailed by increasing the velocity of flow in the air pipe sufficiently to make the H.P. transmitted in both cases the same. It will simplify the calculations to adhere to 44 lbs. as the final indicated pressure, so that we have only to calculate the value of H necessary to make the volume of air transmitted equal to the volume corresponding to a loss of 5 per cent. of the indicated pressure multiplied by the ratios in Columns VII. and VIII. Table VII.

The following Table gives the initial pressures requisite and percentages of loss of head for the sizes of pipes given in Table VII.

TABLE VIII.

Diameter of Pipes.	Air used without Expansion.		Air used with Expansion.	
	Initial Indicated Pressure.	Percentage of Loss.	Initial Indicated Pressure.	Percentage of Loss.
3	53	17	49	10
4	53	17	50	12
5	56	21	51	14
6	56	21	51	14
8	58	23	52	16
10	59	25	52	16
12	60	26	52	16
18	61	27	53	17
24	64	31	55	20

Since the absolute pressures are all in excess of the pressure due to four atmospheres absolute, if Professor Unwin's formula is correct, even with the initial indicated pressures and percentages of losses given in Table VIII., the horse-power which can be transmitted by air is less than half the horse-power which can be transmitted by water at 840 *lbs. initial pressure in pipes of the same size, with a loss of only five per cent. of the pressure.*

In order that the power capable of being transmitted by water at 840 lbs. initial indicated pressure, with a loss of only 5 per cent. of head, may be transmitted by compressed air with the same percentage of loss, in accordance with Box's formula, the initial indicated pressure must be for *non-expansive* working in the case of 3 inch pipes 92 lbs., and of 24-inch pipes 134 lbs., and in accordance with Unwin's, 185 lbs. and 264 lbs. respectively. The corresponding initial pressures for expansive working being, according to Box, 70 lbs. and 110 lbs., and according to Unwin 140 lbs. and 220 lbs. respectively.

Testing the soundness of the joints difficult in the case of air, easy in the case of water.

When water is used for testing the soundness of the joints of a main, the test is applied directly the main is filled with water, and the unsound joints can be at once detected. In the case of air not only is the volume to be pumped equal to R times that of the water, but unless the air during compression and before it enters the pipes is cooled to the temperature of the pipes, directly the pumping ceases the tension of the air will fall owing to the fall in the temperature. Hence, if the test of soundness is the maintenance of a certain degree of pressure for a definite time, the air pumped in ought to be left at such a pressure that the requisite testing pressure will be maintained after the cooling of the air. It is not until after the air has been cooled that the results of the test can be ascertained. In the case of water, the examination of the joints subjected to pressure may be undertaken and completed immediately after they are filled; but in the case of air a time test is necessary, because it is almost impossible to detect by the evidence of sight and sound very small leakages.

Detection of leakage after the commencement of the work of transmitting power.

In the case of water, in ordinary metalled roads the leaks show themselves at once, and when they are laid under concrete, as in the case of asphalted streets, directly the pipe is reached the direction in which the leak must be sought for is shown by the side from which the water leaks into the hole. In the case of asphalted streets, therefore, permanent inspection-places at reasonable intervals will afford the means of detecting the position of the leaks without unnecessary breaking up of the ground.

In the case of air the existence of a leak will rarely be capable of detection, except by the fall in pressure, and its subsequent localisation must be very difficult.

Accumulators of small size sufficient to regulate the flow in the case of high-pressure water, not in case of air of low pressure.

An accumulator, placed at each terminal point of very small capacity, will suffice to render unnecessary any perceptible increase in the average rate of flow during periods of maximum work, when the demand slightly exceeds the average rate of supply, so that the variations in the rate of work will produce very slight variations in the head lost by friction during transmission. The only way in which the same result can be achieved with air of low pressure is to keep the pressure considerably higher than is necessary to do the work, and use also very large terminal receivers.

CHAPTER IV.

SPECIAL CASE OF THE APPLICATION OF COMPRESSED AIR AND HIGH-PRESSURE WATER FOR PUMPING SEWAGE.

Application of compressed air and high-pressure water to pumping.

THE calculated values of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$, when the H.P. refers to the effective power produced for transmission, given in Chapters I. and II., are applicable to all cases, but it was pointed out in those chapters that it is necessary to know the purpose to which the power is to be applied and the mode of its application, before the efficiency of the whole installation can be estimated with any degree of accuracy. When compressed air is used for the purpose of raising liquids, the most economical way of using it is to apply the air direct to the surface of the liquid, since by this means machinery friction is avoided. When water is used for pumping, the simplest machine is a direct-acting pump, in which the slide valves are not *directly* moved by the motion of the piston rod, because in this case, in the act of pumping, the only addition to the net H.P. is the work due to piston friction. The only way to effect this object is to move the slide valve by the direct pressure of the motive water.

Economical and sanitary advantages to be gained by the application of transmitted power for raising sewage.

When transmitted power is used for sewage pumping the area to be drained can be divided into several districts, each being provided with a separate pumping station, to which the motive power is transmitted from one central pumping station. The advantages to be gained by collecting the sewage at several pumping stations are—

(1) Only small pipe sewers at depths sufficient to drain the houses will be necessary.

(2) Whatever the natural configuration of the ground may be, good gradients can always be obtained.

(3) By the use of automatic pumps, which pump the sewage as fast as it flows, the sewers are always kept running free, *and if the automatic action is secured, as it ought to be by the filling of the pump itself, no reservoirs for the storage of the smallest quantity of sewage will be necessary.*

Direct application of compressed air to pumping sewage. Special features.

We have seen in Chapter III. that by properly apportioning the size of the pipes to the power required, this power, whether air or water is used as the medium, can be transmitted in pipes of moderate size a distance of fully 6000 yards, say 4 miles, without entailing a greater loss by pipe friction than 5 per cent. of the initial pressure. In the case of water, the only other points to be considered are leakage and the efficiency of the executive pump, but in the case of air we have also to consider the following special points:—

(1) Possible fall of temperature below the initial temperature of the air before compression.

(2) Effect of variation of lift at the different pumping stations.

(3) Amount of the excess of the maximum over the minimum working pressure necessary to compensate for variation in the flow of the sewage.

Fall in temperature below the initial temperature of the free air.

The initial temperature of the air in the engine room may be taken to vary from about 60° in winter to about 80° in summer, and ultimately in the pipes from about 30° in winter to 50° in summer; or the average fall throughout the year would be about 30°, which corresponds with a fall of from 5·5 to 5·8 per cent. of the initial temperature, so that we must provide for a reduction of fully 5 per cent. of the initial volume of compressed air at initial temperatures.

Effect of the variation in lift at the different pumping stations.

When the air is applied direct to the surface of the sewage, the pressure must be sufficient to raise the sewage from the lowest pumping station, and may therefore be greatly in excess of the pressure required to raise it from the highest. If, therefore, the air is used at the highest station without expansion, not only will the whole of the work of compressing the minimum weight of air required from the pressure necessary at the highest to the pressure necessary at the lowest have to be done, but the weight of air also required will be equal to the minimum weight multiplied by the ratio $\frac{R_1}{R_2}$, R_1 being the ratio of compression at the lowest, and R_2 at the highest station.

The differences in the weight of air required may to a great extent be done away with by expanding the air before it is used by means of an expansion valve, but not wholly, because, owing to the fall in temperature during expansion, its final volume will be to the volume due to isothermal expansion in the ratio of $\left(\frac{R_2}{R_1}\right)^{\frac{\gamma-1}{\gamma}}$. Each particular case must necessarily be taken by itself, but a good idea of the effect of this variation may be obtained by taking the case of only one variation in lift.

Let v_1, v_2 be the minimum volumes of compressed air necessary at the lowest and highest pumping stations to raise the sewage at the absolute pressures $R_1 p_0$ and $R_2 p_0$ respectively, then will the volume of free air required to be pumped be equal to $R_1 v_1$ and $R_2 v_2$. If the volume of sewage at the highest station is lifted by means of air at the pressure $R_1 p_0$, the volume of free air to be pumped will be equal to $R_1 v_2$, and if expanded after compression to $R_2 v_2 \left(\frac{R_1}{R_2}\right)^{\frac{\gamma-1}{\gamma}}$.

For the sake of simplifying the work of comparison, we will take the case of perfect isothermal pumping. The loss of work calculated on this basis will be less than the actual loss sustained in practice, but the investigation will fully illustrate the way in which the power is wasted.

If the air were compressed by two separate engines to the absolute pressures $R_1 p_0$, $R_2 p_0$, the work done in isothermal pumping in the cylinders of the compressor would be equal to

$$p_0 R_1 v_1 \log_e R_1 + p_0 R_2 v_2 \log_e R_2 ; \qquad \text{A.}$$

and it is the ratio of the work done by one compressing engine in the two sets of experiments described to this ratio, which we have to find.

Case I.—Air compressed to the highest pressure used at both pumping stations without expansion.

Since the volume of free air is in this case equal to $R_1 (v_1 + v_2)$, the work done in the air cylinder will be equal to

$$p_0 R_1 (v_1 + v_2) \log_e R_1 ,$$

and the ratio of this to A is

$$\frac{R_1 (v_1 + v_2) \log_e R_1}{R_1 v_1 \log_e R_1 + R_2 v_2 \log_e R_2} ,$$

which reduces to

$$\frac{R_1 (m + 1) \log_e R_1}{R_1 \log_e R_1 + m R_2 \log_e R_2} , \qquad \text{B.}$$

if $v_2 = m v_1$.

Case II.—Air compressed to the highest pressure but expanded before using to the lowest pressure sufficient to raise the sewage at the highest station.

Since the volume of free air required to raise the sewage from the highest pumping station is equal to $R_2 v_2 \left(\frac{R_1}{R_2}\right)^{\frac{\gamma-1}{\gamma}}$, the work to be done in this case will be equal to

$$p_0 \left(R_1 v_1 + R_2 v_2 \left(\frac{R_1}{R_2}\right)^{\frac{\gamma-1}{\gamma}}\right) \log_e R_1,$$

and the ratio of this to A will be

$$\frac{\left(R_1 + m R_2 \left(\frac{R_1}{R_2}\right)^{\frac{\gamma-1}{\gamma}}\right) \log_e R_1}{R_1 \log_e R_1 + m R_2 \log_e R_2}. \qquad \text{C.}$$

For the sake of illustration we may take $R_1 = 2 R_2$ and R_2 equal successively to $1\frac{1}{2}$ and 2. The following Table contains the values of the ratios B and C, corresponding to these two values of R_1 and R_2 for values of m from $\frac{1}{4}$ to 4:—

Value of m.	Minimum Lift 17 ft. Maximum Lift 34 ft.		Minimum Lift 34 ft. Maximum Lift 68 ft.	
	B.	C.	B.	C.
$\frac{1}{4}$	1·19	1·08	1·18	1·09
$\frac{1}{3}$	1·25	1·13	1·23	1·12
$\frac{1}{2}$	1·37	1·19	1·33	1·19
1	1·68	1·35	1·60	1·33
2	2·19	1·63	2·00	1·49
3	2·57	1·80	2·28	1·65
4	2·87	1·92	2·50	1·75

The ratios in column C are those due to *perfect* expansive working, which cannot be achieved in practice, we may therefore regard the mean of the four values in each case as about the correct working value. If we take the case where $m = 1$, or where the same quantities are raised from each lift, as the average one likely to be met with in practice, the mean of the four values is 1·49, so that it will be necessary to add 49 per cent. to the net work done in raising sewage on account of variation of lift.

Excess of the maximum over the minimum working pressure necessary to compensate for the variation in the flow of the sewage.

When compressed air is used for pumping a varying volume of flow of sewage, the additional volume of compressed air required to deal with the excess, over the average rate of flow, can only be secured by compressing the air in the receivers to a higher pressure than would be necessary if the volume to be dealt with per minute were constant, and by providing air-compressors of sufficient capacity to pump the requisite volume for the average rate of flow when running considerably below their maximum efficient speed. When the rate of flow exceeds the average supply the extra volume of air used will cause the pressure to fall in the receiver, and the extra volume will be provided partly by expansion of the air already pumped, partly by the increased speed of the air-pumps consequent on the fall in the pressure. When the increased rate of consumption of air first commences, the pressure of the air will decrease gradually, so that the excess in the volume of sewage raised in the interval, during which the tension of the air falls from the maximum to the minimum pressure, will not be equal to the difference between the volumes due to the two pressures even if no account is taken of the loss of volume due to the fall of temperature during expansion. We may, however, assume for the sake of illustration that the increased speed of the engine makes up the deficiency, and that the extra volume of sewage pumped is, therefore, equal to the difference between the volumes of the air at the higher and lower pressures.

When the sewage has to be pumped as fast as it flows, it will not be safe to estimate the average volume per hour during the periods of maximum flow at less than one-eighth of the daily flow of sewage. During these hours of maximum flow a variation of 25 per cent. above and below the average for a period of half an hour cannot be regarded as excessive. If we adopt these data, the difference between the volume of air in the receiver at minimum and maximum pressure must be equal to 25 per cent. of one half hour's maximum average flow, or to $\frac{1}{64}$th part of the daily volume of flow. For a population of 1000 at 30 gallons a head, this excess over the maximum average will be 470 gallons, say 500 gallons. If V be the volume contained by the receiver in gallons, and P, p, the maximum and minimum air pressures, the difference between the volumes at the two pressures will be equal to $\left(\frac{P}{p} - 1\right)V$, and therefore

$$V = \frac{500\,p}{P - p} = 1500 \text{ gallons}$$

if $p = \cdot 75\,P$, which corresponds with $P = 1 \cdot 34\,p$. In order, therefore, that the maximum pressure necessary to be adopted to keep the sewers always running free should not exceed by more than 34 per cent. the minimum pressure required to raise the sewage, the receiver capacity provided must be at the rate of 1500 gallons for every 1000 population, or for 50,000, the population of an average-sized town, the capacity must be 75,000 gallons.

Efficiency of the executive pump.

When the air is applied directly to the surface of the sewage the only loss in the executive pump will be due to the space occupied by the free air in the pump immediately before the compressed air is admitted. In well-designed pumps of this description the loss due to this cause should not exceed 5 per cent. when the pump works efficiently. In automatic pumps, however, of this description, when the automatic action is regulated by the inflow and outflow of the sewage itself, the alternate admission and exhaust of the compressed air can only be accomplished by means of a float, which rises and falls as the liquid rises and falls in the pump-barrel, attached by means of a piston passing through a stuffing-box to the valve which regulates the times of admission and exhaust. When the packing in the stuffing-box is too slack the valve is liable to act too soon, and to admit the compressed air before the pump-barrel is filled with sewage, and an additional volume of compressed air, varying in amount, is wasted whenever this occurs. It is impossible to make any estimate of the amount thus liable to be wasted. The only way to guard against loss from this cause is to attach a glass gauge to the sides of the pump-barrel, which will show, by the level of the liquid in it at the period of admission, when the valve is working regularly.

Leakage.

Neither in the case of air or water will the amount of leakage be appreciable if the pipes are well jointed and the workmanship of the machines of the best quality.

Summary of the additions to be made to the net work done in raising sewage.

The following summary gives the work which has to be added to the net work done in raising sewage to obtain the value of the effective H.P. to be produced by air pumping in accordance with the above estimates, expressed in terms of the net work done in raising sewage:—

Loss due to pipe friction	·05
„ „ fall of temperature	·05
„ „ variation of lift	·49
„ „ „ flow	·34
„ „ „ pump clearance	·05
Total estimated loss	·98

Leaving, therefore, possible further loss by irregular valve action and leakage out of consideration, the ratios in Table III. must be multiplied by 1·98, say 2, to obtain the safe and working values of the ratios $\frac{\text{I.H.P.}}{\text{H.P.}}$ for the whole installation.

Estimated values of safe and working ratios.

The following table gives the safe and working ratios for the values of R used in the previous chapters. The heights of the lift given in the table are the equivalents of the minimum indicated pressures necessary at the pumps, and therefore only equal to 70 per cent. of the height due to the maximum pressure in the receiver, since 5 per cent. is lost by pipe friction and 25 per cent. is expended in providing for variation in flow. The heights in the table are equal to the sum of the dead lift and the height due to friction in the rising main. The working ratio is the partial isothermal ratio described in Chapter I. The values of the ratios are given to the nearest quarter of an integer.

Value of R	1½	2	3	4	5	6	8	10	15
Height of lift in feet ..	12	23	45	68	90	114	158	204	318
Safe value of $\frac{\text{I.H.P.}}{\text{H.P.}}$ ..	5	6¾	8¼	10½	11¾	13¼	15½	18	22
Working value ditto ..	3	4	4½	5½	6	6½	7½	8½	10

Application of high-pressure water to sewage pumping.

When high-pressure water is used for the transmission of power for pumping sewage, in computing the maximum work to be done by the power conveyed we have only to take account of the pipe friction and the efficiency of the executive pump. Variation of lift is provided for by varying the value of the ratio of the diameter of the power plunger or piston to the pump plunger or piston, so that the power expended at each pumping station is in exact proportion to the work to be done. Variation in the rate of flow can easily be provided for by the use of accumulators in which the pressure remains constant, so that the ratio of $\frac{\text{I.H.P.}}{\text{H.P.}}$ for all variations of flow remains constant. Accumulators, which are simply vertical cylinders provided with movable pistons loaded with weights sufficient to balance the maximum pressure on the under side, can be used in the case of high-pressure water, because the variation in the volume used above and below the average is small. The total volume of high-pressure water is only a small fraction of the volume of the sewage, rarely more than one-twenty-fifth part; so that on an average an accumulator of 20 gallons capacity would suffice to supply power to raise the 500 gallons excess of sewage maximum flow in the case of a population of 1000, and *accumulators capable of holding* 1000 *gallons would suffice in the case of a population of* 50,000.

The accumulator capacity of 1000 gallons in the case of high-pressure water, and 75,000 gallons receiver capacity in the case of air, are necessary to secure the perfect automatic action of the pumping machinery at the pumping station; but as there is always an engine-driver in charge, who can vary the rate of pumping according to the rate of flow, probably in actual practice accumulator capacity of 100 gallons for water, and receiver capacity of 20,000 gallons for air, would suffice for a population of 50,000.

Ratio of diameter of power-plunger to pump-plunger.

The effective total pressure on the power plunger will be equal to the gross total pressure *minus* the frictional resistance of the plunger packing, and the pressure to be overcome will be equal to the total pressure on the pump plunger *plus* the frictional resistance of the plunger packing (see Chapter II.). In estimating the total pressure on the pump plunger, the pressure per unit of area must be the sum of the pressures due to the dead lift and to rising main friction. If the pump is single acting and the return stroke effected by means of the pressure of the sewage on the under side of the pump piston, or by weights, the equivalent pressure per unit of area must also be added to the above.

F 2

Let

P = pressure of motive fluid in pump barrel.
p = pressure in pump barrel due to all causes described above.
D = diameter of power plunger.
d = " " pump "

Then we must have

$$\cdot 78\, P\, D^2\left(1 - \frac{C}{D}\right) = \cdot 78\, p\, d^2\left(1 + \frac{C}{d}\right)$$

and

$$D = \frac{C}{2} + \sqrt{\frac{C^2}{4} + \frac{pd(d+C)}{P}}.$$

Using for C the value ·6, corresponding with the results for Messrs. Hicks's experiments with new unused leathers, we get

$$D = \cdot 3 + \sqrt{\cdot 09 + \frac{pd(d+\cdot 6)}{P}}. \qquad \text{A.}$$

For the sake of illustration we may adopt 18 inches for the value of d. We may also assume $\frac{P}{p} = 25$ as an average value of the ratio, when P varies from 800 to 900 lbs., and $\frac{P}{p} = 5$ when P varies from 50 to 100 lbs.

$$(1) \quad \frac{P}{p} = 25.$$

From A

$$D = 4 \text{ inches};$$

if there were no frictional resistance we should have

$$D = 3\cdot 6 \text{ inches}.$$

$$(2) \quad \frac{P}{p} = 5.$$

From A

$$D = 8 \cdot 5 \text{ inches};$$

if there were no frictional resistance we should have

$$D = 8 \text{ inches}.$$

With well-used leathers, for which, according to Messrs. Hicks's experiments, $C = \cdot 4$, the diameters need not exceed $3\frac{7}{8}$ inches and $8\frac{1}{3}$ inches respectively.

The total work done by the high-pressure water is equal to $\cdot 78\,P\,D^2$, and the effective work done to $\cdot 78\,p\,d^2$, therefore the ratio of the effective work done to the total work done will be equal to

Efficiency of a pump in which the valve is moved by the direct application of the high-pressure water.

$$\frac{p\,d^2}{P\,D^2} = \frac{p\,d^2}{P\left\{\cdot 3 + \sqrt{\cdot 09 + \frac{p\,d\,(d + \cdot 6)}{P}}\right\}^2}.$$

The following table gives the value of this ratio for pump barrels varying from 9 to 30 inches diameter for values of the ratio $\frac{P}{p}$ varying from 4 to 100, subject to the condition that the power plunger is not to be less than 2 inches diameter.

Value of $P \div p$.	Diameter of Pump Barrel.						
	9	12	15	18	21	24	30
4	·82	·85	·88	·91	·92	·93	·94
10	·76	·81	·85	·88	·90	·91	·93
20	·70	·77	·81	·84	·86	·88	·91
40	..	·71	·75	·80	·81	·84	·87
60	..	..	·71	·76	·78	·81	·85
80	..	..	..	·73	·76	·80	·82
100	..	..	..	..	·74	·77	·80

In average installations the size of the pump barrels will vary from 15 inches to 24 inches, and the maximum pressure on the accumulators can easily be arranged so that the average value of the ratio $P \div p$ does not exceed 20 : 1, and under these conditions the average coefficient of efficiency for the four sizes of pumps will be ·85. When the average value of the ratio of $P \div p$ does not exceed 4 : 1, the average coefficient of efficiency of the four sizes will be ·91.

Height due to velocity in inlet valve portholes.

In discussing the question of compressed air, no account was taken of the loss of head due to this cause, and in the case of water the loss is practically inappreciable. Even with the lowest pressures the area of the inlet porthole need not be less than one-fourth of the area of the power plunger. In pumps of this class the length of stroke ought to be very little more than the diameter of pump-piston, and the number of strokes should not exceed twelve per minute, so that the inlet porthole velocity need never exceed 12 feet per second. The head lost, therefore, due to this cause need never exceed 3 feet.

Volume of water required to move the valve expressed as a percentage of the volume used at each stroke of the pump.

The volume of water required to move the valve varies with the area of the valve-piston and the length of its travel. Both these quantities vary with the size of the power plunger of the pump. Subject to the condition that the velocity on the inlet porthole of the valve is not to exceed sixteen times the velocity of the pump-piston, the volume used need not, even in the case of the lowest pressures, exceed four per cent. of the volume used for pumping.

Safe and working values of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$.

It is abundantly clear from the foregoing investigations that the executive pumping apparatus can be devised to give out an efficiency of fully 80 per cent. of the power transmitted, and that the loss due to transmission need not exceed 5 per cent., so that the efficiency of the installation for utilising the power can be made equal to fully 75 per cent. Following, therefore, the rule we adopted in the case of air, viz. to double the H.P. produced to get the safe value of the ratio $\frac{\text{I.H.P.}}{\text{H.P.}}$, and to add 25 per cent. for machinery friction to obtain the minimum working value, we get, when H.P. means the actual work done in raising sewage, the following values :—

$$\text{Safe value of } \frac{\text{I.H.P}}{\text{H.P.}} = 2\tfrac{1}{2}.$$

$$\text{Minimum working do.} = 1\tfrac{1}{2}.$$

In the case of high-pressure water, special installations of power-producing machinery not always necessary.

When the water supply of any district is in the hands of the local authority, the pumps, when only a small portion of the sewage of any given area has to be pumped, can be economically actuated by water supplied from the street mains at the actual cost of pumping, which would not, in the most extreme cases, exceed 1*d*. per thousand gallons. When the whole of the sewage has to be pumped, special installations of power-producing machinery are of course absolutely indispensable.

Sanitary advantages obtained by the use of high-pressure water instead of compressed air.

The high-pressure water used for actuating the automatic pumps never comes in contact with the sewage, and can never, therefore, entail the slightest risk of spreading any infectious disease. The water after use will be perfectly pure, and may be used for drinking purposes if of suitable quality. It can always be used for flushing the side channels of the streets.

When air is applied directly to the surface of the sewage in ordinary circumstances, no evil results probably need be apprehended during periods of maximum flow, when the air is discharged in the space of half a minute; but when the rate of flow of the sewage is at a minimum, the air in the pump will remain for a long time in contact with the decomposed sewage slime with which the pump is internally coated. If, therefore, there are any germs of disease in the sewage, the air must necessarily become charged with them. The use of ventilating pipes in this case, discharging the exhaust air above the tops of the houses, can at best only free the immediate neighbourhood from the possible consequences of germ-carried disease. The arguments founded on this possibility against the system of water-carried sewage clearly apply much more strongly against the system of raising sewage by compressed air, since the volume of air brought into contact with the surface of the sewage and of the slime-lined surface of the pumps is equal to the volume of the sewage multiplied by the absolute ratio of the air pressure.

CHAPTER V.

COMPARISON OF THE TWO SYSTEMS IN POINT OF EFFICIENCY AND COST OF INSTALLATION.

THE investigations in Chapter III. prove conclusively that the diameters of pipes necessary to convey any assigned quantity of power in the case of water at a pressure of about 800 lbs. are very much less than those required to convey the same quantity of power by means of compressed air, when the pressure to which the air is subjected does not exceed the limits usually found in practice. The extra thickness of the metal of the pipes in the case of high-pressure water is amply compensated by the greater size of the pipes required in the case of air and the greater cost of testing the soundness of the work. We may therefore fairly consider that the cost of the pipes for transmitting the power will be about the same in each case. Cost of transmission.

Since the engines required to utilise water of 800 lbs. indicated pressure are necessarily of much less size than engines of equal I.H.P. actuated by air of 45 lbs. indicated pressure, the first cost of such engines ought to be less. We have seen in Chapter I. that engines for utilising compressed air must be capable of admitting the air during the whole stroke, and therefore ordinary stationary engines cannot be used until their valve gear has been altered. The great advantages obtained from the use of air are that it can be used in any situation, and after exhaustion will serve for ventilation purposes. In Executive engines.

the case of water, difficulties connected with the disposal of the exhaust water will not unfrequently arise.

Comparison of the cost of the installations for producing power in the case of air and water.

The first cost of the prime motors, the engines and boilers, will vary with the I.H.P. required to produce the assigned effective power. The high-pressure water pumps, being of much less size than the air pumps, will cost very much less; and the accumulators, although much more costly than receivers of the same capacity, will probably cost less than the receivers, because the capacity of the latter must be about twenty times that of the accumulators. In the case of compressed air, the buildings required will be much larger also. Taking every point into consideration, we shall clearly not err in this matter in favour of high-pressure water if we estimate the cost of the installation in each case to be in the respective ratios of the safe I.H.P. required.

In the case of sewage we get the following values of the ratios

$$\frac{\text{I.H.P. required in the case of compressed air}}{\text{I.H.P. required in the case of water at 800 lbs. indicated pressure}}.$$

When

$R = 1\frac{1}{2}$	$2 : 1$
$R = 2$	$2\frac{3}{4} : 1$
$R = 3$	$3\frac{1}{2} : 1$
$R = 4$	$4\frac{1}{4} : 1$
$R = 5$	$4\frac{3}{4} : 1$
$R = 6$	$5\frac{1}{2} : 1$
$R = 8$	$6\frac{1}{4} : 1$
$R = 10$	$7\frac{1}{4} : 1$
$R = 15$	$9 : 1$

Comparison of the efficiency.

Conversely, the efficiency of installations of high-pressure water will be to those of compressed air in the inverse ratio of safe I.H.P. required, so that the ratios just given will represent the value of the ratio

$$\frac{\text{Efficiency of high-pressure water}}{\text{Efficiency of compressed air}}.$$

APPENDIX.

LYMINGTON SEWERAGE.

Description of an Installation of Automatic Pumps, the Invention of the Author.

The main sewerage of this town is now being carried out in accordance with the plans of Mr. James Lemon, M. Inst. C.E., the principle adopted being that of discharging the sewage, in its crude state, into the Solent through an iron outfall pipe of about 2000 yards in length across the foreshore. The larger part of the district will be drained by gravitation, but there is a small portion of the town, adjacent to the river, which is very low, and in order to secure a constant flow in the sewers it will be pumped by one of Donaldson's patent automatic pumps into an iron main, which will discharge itself at the head of the outfall sewer. In the original scheme Mr. Lemon proposed to pump the low-level sewage by means of two small centrifugal pumps, driven by Crossley's Otto gas-engines in duplicate, but on reconsideration he decided to adopt the automatic pumps before referred to. In a small district like this it is important that the working expenses should be kept down as low as possible, more especially the cost of labour. Gas-engines, as is well known, will work without much attention, but still they require oiling and cleaning, and occasionally looking to, in order to secure their efficient working. But by the automatic pumps this is avoided, as they will work the same as an ejector, and there are no movable parts requiring oiling, except the journals of the wheels which support the balance weights. The engineer also considered the desirability of pumping the sewage by means of two of Shone's ejectors; but the great distance of the pumps from the waterworks, where the power would be generated, involving a very long length of air main, would have increased the first cost of the works so materially as to render their adoption, on the ground of expense, out of the question. The automatic pumps being worked by the pressure of water from the main, it is only a question of pumping so much more water at the waterworks, which are the property of the Local Board. In addition to ensuring a constant flow in the low-level district, these pumps will also be the means of flushing the main outfall, as they will lift the sewage above the level of high-water, and so put the outfall under pressure. There are also means of flushing the outfall by a large storage reservoir sewer in which the sewage will occasionally be headed up and discharged at convenient times of the tide.

The pump chamber is about 13 feet below the level of high water, and as it is constructed near the river bank, special means have been adopted to render it water-tight. It is built of brickwork in cement rendered with 1 inch of Portland cement all round, encased with Portland cement concrete and clay puddle, 12 inches in thickness, and faced on the inside with white bricks, so as to render the chamber as light as possible. The chamber in which the pumps are placed is 15 feet by 10 feet by 17 feet high, and as it is only a few feet above the surface of the ground, it will not obstruct the view of the houses in the vicinity. In the original scheme, objections were raised by the owners and occupiers of property to the proposed engine-house, which have been removed by the proposed chambers. Adjoining the pump chamber there are two screen chambers, which will prevent any large floating substances finding their way into the pumps. Four penstocks are provided for shutting off the sewage at will, in case of repairs or for other causes.

The chamber is covered by Hyatt's patent lens-lights, supported by rolled iron girders, and a platform is provided of iron plates for the purpose of inspecting the machinery.

The cost of the pump-chamber and pumps will be about 100*l.* less than the estimated cost of the gas engines and engine house, and there will be a saving on the working expenses, in favour of the automatic pumps, of about 70*l.* per annum.

Sheet No. 1 gives the details of the pump-house erected for the two automatic pumps, and Sheet No. 2 the details of the construction of the pump.

The balance weights are so adjusted that the upward pressure of the sewage on the under side of the pump piston is sufficient to raise the piston to its highest position, when the sewage in the inlet down pipe is on a level with the invert of the horizontal inlet pipe, so that the alternate action of the pumps is rendered certain, because the first pump must be filled before much sewage can reach the second.

It will be seen from the detail drawings that there are no tappets to work the valve, which is moved by the alternate admission and exhaust of the high-pressure water to the opposite ends of the valve cylinder, effected by means of the passages shown on the power plunger.

The automatic action is secured in two ways, first by the use of pressure on the under side of the piston to raise it, secondly by the use of an inverted syphon for the lowest portion of the inlet pipe, the bend of which must be placed a few inches below the bottom of the pump piston in its lowest position. Directly the sewage in the inlet pipe falls below the level of the bottom of the pump piston, the upward motion will necessarily cease, even if the weights of themselves were sufficient to raise the pump piston without the help of the upward pressure of the sewage on the under side of the piston.

The pumps have each to raise 194 gallons per minute with 50 lbs. pressure of water to a height, including that due to friction, of 20 feet above the invert of the sewer. The pumps are each 18 inches diameter and 21 inches stroke, the power plunger of one pump being 9 inches and of the other 9½ inches diameter. The bottom of the pump piston in its lowest position will be 2 feet 8 inches, and in its highest 11 inches below the invert of the sewer. The exhaust water has to be delivered to a height of 10 feet above the invert of the sewer. With water of several hundred pounds pressure the power cylinders would be of small diameter, and therefore the height at which the exhaust water would ordinarily have to be delivered need not in that case be taken into consideration, but as the plungers are in this case 9 and 9½ inches in diameter, an addition of from 60 to 80 lbs. to the balance-weights is needed to raise the exhaust water. This would be equivalent to a pressure of about ·3 lb. on the under side of the pump piston, so that the resistance to be overcome per unit of area would be as follows:—

Pressure due to 20 ft. lift	8·8 lbs. per sq. in.		
Pressure due to depth of bottom of pump piston below invert 2 ft. 8 in.	1·2	„	„
Pressure equivalent to increase of weights ..	·3	„	„
Total	10·3	„	„

Practically, therefore, we may consider the ratio of the pressure in the power cylinder to that in the pump barrel to be 5 : 1, and the minimum size of the power plunger would be, in accordance with the formula given in Chapter IV., 8½ inches. As the motive water is supplied from the mains in the town, where the pressure is necessarily subject to great variations, the power plunger of the first pump has been made 9 inches, and of the second, or reserve, pump 9½ inches. No appreciable sacrifice of efficiency of the installation as a whole is caused by this increase in the size of the plunger of the second pump, because nearly the whole of the work will be done by the pump with the 9-inch plunger, which is nearest to the sewer. A pressure of about 40 lbs. to the square inch will suffice to work the pump with the 9½-inch plunger.

LONDON: PRINTED BY WILLIAM CLOWES AND SONS, LIMITED, STAMFORD STREET AND CHARING CROSS.

WORKS BY THE SAME AUTHOR.

A TREATISE ON THE

ART OF CONSTRUCTING OBLIQUE ARCHES, WITH SPIRAL COURSES.

Plates, 8*vo*, *cloth*, 4*s*.

SWITCHES AND CROSSINGS;

FORMULÆ FOR ASCERTAINING THE ANGLES OF CROSSINGS, THE LENGTHS OF SWITCHES, AND THE DISTANCES OF THE POINTS OF THE CROSSINGS AND THE HEELS OF THE SWITCHES FROM THE SPRINGING OF THE CURVE.

Plates, 8*vo*, *cloth*, 10*s*. 6*d*.

TABLES FOR PLATELAYERS,

COMPILED FROM THE FORMULÆ IN THE WORK ON SWITCHES AND CROSSINGS.

Plates, 12*mo*, *cloth*, 3*s*. 6*d*.

NEW FORMULAS FOR THE

LOADS AND DEFLECTIONS OF SOLID BEAMS AND GIRDERS.

8*vo*, *cloth*, 4*s*. 6*d*.

PRINCIPLES OF

CONSTRUCTION AND EFFICIENCY OF WATER WHEELS.

8*vo*, *cloth*, 5*s*.

DONALDSON'S

PONCELET TURBINE AND WATER PRESSURE ENGINE AND PUMP.

PREFACED BY A SHORT TREATISE ON THE IMPULSIVE ACTION OF INELASTIC FLUIDS.

4*to*, *cloth*, 5*s*.

E. & F. N. SPON, 125, STRAND, LONDON.

BOOKS RELATING

TO

APPLIED SCIENCE,

PUBLISHED BY

E. & F. N. SPON,

LONDON: 125, STRAND.

NEW YORK: 12, CORTLANDT STREET.

A Pocket-Book for Chemists, Chemical Manufacturers, Metallurgists, Dyers, Distillers, Brewers, Sugar Refiners, Photographers, Students, etc., etc. By THOMAS BAYLEY, Assoc. R.C. Sc. Ireland, Analytical and Consulting Chemist and Assayer. Fourth edition, with additions, 437 pp., royal 32mo, roan, gilt edges, 5*s.*

SYNOPSIS OF CONTENTS:

Atomic Weights and Factors—Useful Data—Chemical Calculations—Rules for Indirect Analysis—Weights and Measures—Thermometers and Barometers—Chemical Physics—Boiling Points, etc.—Solubility of Substances—Methods of Obtaining Specific Gravity—Conversion of Hydrometers—Strength of Solutions by Specific Gravity—Analysis—Gas Analysis—Water Analysis—Qualitative Analysis and Reactions—Volumetric Analysis—Manipulation—Mineralogy—Assaying—Alcohol—Beer—Sugar—Miscellaneous Technological matter relating to Potash, Soda, Sulphuric Acid, Chlorine, Tar Products, Petroleum, Milk, Tallow, Photography, Prices, Wages, Appendix, etc., etc.

The Mechanician: A Treatise on the Construction and Manipulation of Tools, for the use and instruction of Young Engineers and Scientific Amateurs, comprising the Arts of Blacksmithing and Forging; the Construction and Manufacture of Hand Tools, and the various Methods of Using and Grinding them; the Construction of Machine Tools, and how to work them; Machine Fitting and Erection; description of Hand and Machine Processes; Turning and Screw Cutting; principles of Constructing and details of Making and Erecting Steam Engines, and the various details of setting out work, etc., etc. By CAMERON KNIGHT, Engineer. *Containing* 1147 *illustrations*, and 397 pages of letter-press, Fourth edition, 4to, cloth, 18*s.*

Just Published, in Demy 8vo, cloth, containing 975 pages and 250 Illustrations, price 7s. 6d.

SPONS' HOUSEHOLD MANUAL:

A Treasury of Domestic Receipts and Guide for Home Management.

PRINCIPAL CONTENTS.

Hints for selecting a good House, pointing out the essential requirements for a good house as to the Site, Soil, Trees, Aspect, Construction, and General Arrangement; with instructions for Reducing Echoes, Waterproofing Damp Walls, Curing Damp Cellars.

Sanitation.—What should constitute a good Sanitary Arrangement; Examples (with illustrations) of Well- and Ill-drained Houses; How to Test Drains; Ventilating Pipes, etc.

Water Supply.—Care of Cisterns; Sources of Supply; Pipes; Pumps; Purification and Filtration of Water.

Ventilation and Warming.—Methods of Ventilating without causing cold draughts, by various means; Principles of Warming; Health Questions; Combustion; Open Grates; Open Stoves; Fuel Economisers; Varieties of Grates; Close-Fire Stoves; Hot-air Furnaces; Gas Heating; Oil Stoves; Steam Heating; Chemical Heaters; Management of Flues; and Cure of Smoky Chimneys.

Lighting.—The best methods of Lighting; Candles, Oil Lamps, Gas, Incandescent Gas, Electric Light; How to test Gas Pipes; Management of Gas.

Furniture and Decoration.—Hints on the Selection of Furniture; on the most approved methods of Modern Decoration; on the best methods of arranging Bells and Calls; How to Construct an Electric Bell.

Thieves and Fire.—Precautions against Thieves and Fire; Methods of Detection; Domestic Fire Escapes; Fireproofing Clothes, etc.

The Larder.—Keeping Food fresh for a limited time; Storing Food without change, such as Fruits, Vegetables, Eggs, Honey, etc.

Curing Foods for lengthened Preservation, as Smoking, Salting, Canning, Potting, Pickling, Bottling Fruits, etc.; Jams, Jellies, Marmalade, etc.

The Dairy.—The Building and Fitting of Dairies in the most approved modern style; Butter-making; Cheesemaking and Curing.

The Cellar.—Building and Fitting; Cleaning Casks and Bottles; Corks and Corking; Aërated Drinks; Syrups for Drinks; Beers; Bitters; Cordials and Liqueurs; Wines; Miscellaneous Drinks.

The Pantry.—Bread-making; Ovens and Pyrometers; Yeast; German Yeast; Biscuits; Cakes; Fancy Breads; Buns.

The Kitchen.—On Fitting Kitchens; a description of the best Cooking Ranges, close and open; the Management and Care of Hot Plates, Baking Ovens, Dampers, Flues, and Chimneys; Cooking by Gas; Cooking by Oil; the Arts of Roasting, Grilling, Boiling, Stewing, Braising, Frying.

Receipts for Dishes—Soups, Fish, Meat, Game, Poultry, Vegetables, Salads, Puddings, Pastry, Confectionery, Ices, etc., etc.; Foreign Dishes.

The Housewife's Room.—Testing Air, Water, and Foods; Cleaning and Renovating; Destroying Vermin.

Housekeeping, Marketing.

The Dining-Room.—Dietetics; Laying and Waiting at Table; Carving; Dinners, Breakfasts, Luncheons, Teas, Suppers, etc.

The Drawing-Room.—Etiquette; Dancing; Amateur Theatricals; Tricks and Illusions; Games (indoor).

The Bedroom and Dressing-Room; Sleep; the Toilet; Dress; Buying Clothes; Outfits; Fancy Dress.

The Nursery.—The Room; Clothing; Washing; Exercise; Sleep; Feeding; Teething; Illness; Home Training.

The Sick-Room.—The Room; the Nurse; the Bed; Sick Room Accessories; Feeding Patients; Invalid Dishes and Drinks; Administering Physic; Domestic Remedies; Accidents and Emergencies; Bandaging; Burns; Carrying Injured Persons; Wounds; Drowning; Fits; Frost-bites; Poisons and Antidotes; Sunstroke; Common Complaints; Disinfection, etc.

The Bath-Room.—Bathing in General; Management of Hot-Water System.

The Laundry.—Small Domestic Washing Machines, and methods of getting up linen; Fitting up and Working a Steam Laundry.

The School-Room.—The Room and its Fittings; Teaching, etc.

The Playground.—Air and Exercise; Training; Outdoor Games and Sports.

The Workroom.—Darning, Patching, and Mending Garments.

The Library.—Care of Books.

The Garden.—Calendar of Operations for Lawn, Flower Garden, and Kitchen Garden.

The Farmyard.—Management of the Horse, Cow, Pig, Poultry, Bees, etc., etc.

Small Motors.—A description of the various small Engines useful for domestic purposes, from 1 man to 1 horse power, worked by various methods, such as Electric Engines, Gas Engines, Petroleum Engines, Steam Engines, Condensing Engines, Water Power, Wind Power, and the various methods of working and managing them.

Household Law.—The Law relating to Landlords and Tenants, Lodgers, Servants, Parochial Authorities, Juries, Insurance, Nuisance, etc.

On Designing Belt Gearing. By E. J. COWLING WELCH, Mem. Inst. Mech. Engineers, Author of 'Designing Valve Gearing.' Fcap. 8vo, sewed, 6*d.*

A Handbook of Formulæ, Tables, and Memoranda, for Architectural Surveyors and others engaged in Building. By J. T. HURST, C.E. Fourteenth edition, royal 32mo, roan, 5*s.*

"It is no disparagement to the many excellent publications we refer to, to say that in our opinion this little pocket-book of Hurst's is the very best of them all, without any exception. It would be useless to attempt a recapitulation of the contents, for it appears to contain almost *everything* that anyone connected with building could require, and, best of all, made up in a compact form for carrying in the pocket, measuring only 5 in. by 3 in., and about ¼ in. thick, in a limp cover. We congratulate the author on the success of his laborious and practically compiled little book, which has received unqualified and deserved praise from every professional person to whom we have shown it."—*The Dublin Builder.*

Tabulated Weights of Angle, Tee, Bulb, Round, Square, and Flat Iron and Steel, and other information for the use of Naval Architects and Shipbuilders. By C. H. JORDAN, M.I.N.A. Fourth edition, 32mo, cloth, 2*s.* 6*d.*

A Complete Set of Contract Documents for a Country Lodge, comprising Drawings, Specifications, Dimensions (for quantities), Abstracts, Bill of Quantities, Form of Tender and Contract, with Notes by J. LEANING, printed in facsimile of the original documents, on single sheets fcap., in paper case, 10*s.*

A Practical Treatise on Heat, as applied to the Useful Arts; for the Use of Engineers, Architects, &c. By THOMAS BOX. *With* 14 *plates.* Third edition, crown 8vo, cloth, 12*s.* 6*d.*

A Descriptive Treatise on Mathematical Drawing Instruments: their construction, uses, qualities, selection, preservation, and suggestions for improvements, with hints upon Drawing and Colouring. By W. F. STANLEY, M.R.I. Fifth edition, *with numerous illustrations,* crown 8vo, cloth, 5*s.*

Quantity Surveying. By J. LEANING. With 42 illustrations. Second edition, revised, crown 8vo, cloth, 9*s.*

CONTENTS :

A complete Explanation of the London Practice.
General Instructions.
Order of Taking Off.
Modes of Measurement of the various Trades.
Use and Waste.
Ventilation and Warming.
Credits, with various Examples of Treatment.
Abbreviations.
Squaring the Dimensions.
Abstracting, with Examples in illustration of each Trade.
Billing.
Examples of Preambles to each Trade.
Form for a Bill of Quantities.
Do. Bill of Credits.
Do. Bill for Alternative Estimate.
Restorations and Repairs, and Form of Bill.
Variations before Acceptance of Tender.
Errors in a Builder's Estimate.
Schedule of Prices.
Form of Schedule of Prices.
Analysis of Schedule of Prices.
Adjustment of Accounts.
Form of a Bill of Variations.
Remarks on Specifications.
Prices and Valuation of Work, with Examples and Remarks upon each Trade.
The Law as it affects Quantity Surveyors, with Law Reports.
Taking Off after the Old Method.
Northern Practice.
The General Statement of the Methods recommended by the Manchester Society of Architects for taking Quantities.
Examples of Collections.
Examples of "Taking Off" in each Trade.
Remarks on the Past and Present Methods of Estimating.

Spons' Architects' and Builders' Price Book, with useful Memoranda. Edited by W. YOUNG, Architect. Crown 8vo, cloth, red edges, 3*s.* 6*d.* *Published annually.* Sixteenth edition. *Now ready.*

Long-Span Railway Bridges, comprising Investigations of the Comparative Theoretical and Practical Advantages of the various adopted or proposed Type Systems of Construction, with numerous Formulæ and Tables giving the weight of Iron or Steel required in Bridges from 300 feet to the limiting Spans; to which are added similar Investigations and Tables relating to Short-span Railway Bridges. Second and revised edition. By B. BAKER, Assoc. Inst. C.E. *Plates*, crown 8vo, cloth, 5*s.*

Elementary Theory and Calculation of Iron Bridges and Roofs. By AUGUST RITTER, Ph.D., Professor at the Polytechnic School at Aix-la-Chapelle. Translated from the third German edition, by H. R. SANKEY, Capt. R.E. With 500 *illustrations*, 8vo, cloth, 15*s.*

The Elementary Principles of Carpentry. By THOMAS TREDGOLD. Revised from the original edition, and partly re-written, by JOHN THOMAS HURST. Contained in 517 pages of letter-press, and *illustrated with* 48 *plates and* 150 *wood engravings.* Sixth edition, reprinted from the third, crown 8vo, cloth, 12*s.* 6*d.*

Section I. On the Equality and Distribution of Forces—Section II. Resistance of Timber—Section III. Construction of Floors—Section IV. Construction of Roofs—Section V. Construction of Domes and Cupolas—Section VI. Construction of Partitions—Section VII. Scaffolds, Staging, and Gantries—Section VIII. Construction of Centres for Bridges—Section IX. Coffer-dams, Shoring, and Strutting—Section X. Wooden Bridges and Viaducts—Section XI. Joints, Straps, and other Fastenings—Section XII. Timber.

The Builder's Clerk: a Guide to the Management of a Builder's Business. By THOMAS BALES. Fcap. 8vo, cloth, 1*s.* 6*d.*

Our Factories, Workshops, and Warehouses: their Sanitary and Fire-Resisting Arrangements. By B. H. THWAITE, Assoc. Mem. Inst. C.E. *With* 183 *wood engravings*, crown 8vo, cloth, 9*s.*

Gold: Its Occurrence and Extraction, embracing the Geographical and Geological Distribution and the Mineralogical Characters of Gold-bearing rocks; the peculiar features and modes of working Shallow Placers, Rivers, and Deep Leads; Hydraulicing; the Reduction and Separation of Auriferous Quartz; the treatment of complex Auriferous ores containing other metals; a Bibliography of the subject and a Glossary of Technical and Foreign Terms. By ALFRED G. LOCK, F.R.G.S. *With numerous illustrations and maps*, 1250 pp., super-royal 8vo, cloth, 2*l.* 12*s.* 6*d.*

Iron Roofs: Examples of Design, Description. *Illustrated with* 64 *Working Drawings of Executed Roofs.* By ARTHUR T. WALMISLEY, Assoc. Mem. Inst. C.E. Second edition, revised, imp. 4to, half-morocco, 3*l.* 3*s.*

A History of Electric Telegraphy, to the Year 1837. Chiefly compiled from Original Sources, and hitherto Unpublished Documents, by J. J. FAHIE, Mem. Soc. of Tel. Engineers, and of the International Society of Electricians, Paris. Crown 8vo, cloth, 9*s.*

Spons' Information for Colonial Engineers. Edited by J. T. HURST. Demy 8vo, sewed.

No. 1, Ceylon. By ABRAHAM DEANE, C.E. 2*s.* 6*d.*

CONTENTS:

Introductory Remarks — Natural Productions — Architecture and Engineering — Topography, Trade, and Natural History—Principal Stations—Weights and Measures, etc., etc.

No. 2. Southern Africa, including the Cape Colony, Natal, and the Dutch Republics. By HENRY HALL, F.R.G.S., F.R.C.I. With Map. 3*s.* 6*d.*

CONTENTS:

General Description of South Africa—Physical Geography with reference to Engineering Operations—Notes on Labour and Material in Cape Colony—Geological Notes on Rock Formation in South Africa—Engineering Instruments for Use in South Africa—Principal Public Works in Cape Colony: Railways, Mountain Roads and Passes, Harbour Works, Bridges, Gas Works, Irrigation and Water Supply, Lighthouses, Drainage and Sanitary Engineering, Public Buildings, Mines—Table of Woods in South Africa—Animals used for Draught Purposes—Statistical Notes—Table of Distances—Rates of Carriage, etc.

No. 3. India. By F. C. DANVERS, Assoc. Inst. C.E. With Map. 4*s.* 6*d.*

CONTENTS:

Physical Geography of India—Building Materials—Roads—Railways—Bridges—Irrigation — River Works — Harbours — Lighthouse Buildings — Native Labour — The Principal Trees of India—Money—Weights and Measures—Glossary of Indian Terms, etc.

A Practical Treatise on Coal Mining. By GEORGE G. ANDRÉ, F.G.S., Assoc. Inst. C.E., Member of the Society of Engineers. *With* 82 *lithographic plates.* 2 vols., royal 4to, cloth, 3*l.* 12*s.*

A Practical Treatise on Casting and Founding, including descriptions of the modern machinery employed in the art. By N. E. SPRETSON, Engineer. Third edition, with 82 *plates* drawn to scale, 412 pp., demy 8vo, cloth, 18*s.*

The Depreciation of Factories and their Valuation. By EWING MATHESON, M. Inst. C.E. 8vo, cloth, 6*s.*

A Handbook of Electrical Testing. By H. R. KEMPE, M.S.T.E. Fourth edition, revised and enlarged, crown 8vo, cloth, 16*s.*

Gas Works: their Arrangement, Construction, Plant, and Machinery. By F. COLYER, M. Inst. C.E. *With* 31 *folding plates,* 8vo, cloth, 24*s.*

The Clerk of Works: a Vade-Mecum for all engaged in the Superintendence of Building Operations. By G. G. HOSKINS, F.R.I.B.A. Third edition, fcap. 8vo, cloth, 1*s.* 6*d.*

American Foundry Practice: Treating of Loam, Dry Sand, and Green Sand Moulding, and containing a Practical Treatise upon the Management of Cupolas, and the Melting of Iron. By T. D. WEST, Practical Iron Moulder and Foundry Foreman. Second edition, *with numerous illustrations,* crown 8vo, cloth, 10*s.* 6*d.*

The Maintenance of Macadamised Roads. By T. CODRINGTON, M.I.C.E, F.G.S., General Superintendent of County Roads for South Wales. 8vo, cloth, 6*s.*

Hydraulic Steam and Hand Power Lifting and Pressing Machinery. By FREDERICK COLYER, M. Inst. C.E., M. Inst. M.E. *With* 73 *plates,* 8vo, cloth, 18*s.*

Pumps and Pumping Machinery. By F. COLYER, M.I.C.E., M.I.M.E. *With* 23 *folding plates,* 8vo, cloth, 12*s.* 6*d.*

Pumps and Pumping Machinery. By F. COLYER. Second Part. *With* 11 *large plates,* 8vo, cloth, 12*s.* 6*d.*

A Treatise on the Origin, Progress, Prevention, and Cure of Dry Rot in Timber; with Remarks on the Means of Preserving Wood from Destruction by Sea-Worms, Beetles, Ants, etc. By THOMAS ALLEN BRITTON, late Surveyor to the Metropolitan Board of Works, etc., etc. *With* 10 *plates,* crown 8vo, cloth, 7*s.* 6*d.*

The Municipal and Sanitary Engineer's Handbook. By H. PERCY BOULNOIS, Mem. Inst. C.E., Borough Engineer, Portsmouth. *With numerous illustrations*, demy 8vo, cloth, 12*s.* 6*d.*

CONTENTS:

The Appointment and Duties of the Town Surveyor—Traffic—Macadamised Roadways—Steam Rolling—Road Metal and Breaking—Pitched Pavements—Asphalte—Wood Pavements—Footpaths—Kerbs and Gutters—Street Naming and Numbering—Street Lighting—Sewerage—Ventilation of Sewers—Disposal of Sewage—House Drainage—Disinfection—Gas and Water Companies, etc., Breaking up Streets—Improvement of Private Streets—Borrowing Powers—Artizans' and Labourers' Dwellings—Public Conveniences—Scavenging, including Street Cleansing—Watering and the Removing of Snow—Planting Street Trees—Deposit of Plans—Dangerous Buildings—Hoardings—Obstructions—Improving Street Lines—Cellar Openings—Public Pleasure Grounds—Cemeteries—Mortuaries—Cattle and Ordinary Markets—Public Slaughter-houses, etc.—Giving numerous Forms of Notices, Specifications, and General Information upon these and other subjects of great importance to Municipal Engineers and others engaged in Sanitary Work.

Metrical Tables. By G. L. MOLESWORTH, M.I.C.E. 32mo, cloth, 1*s.* 6*d.*

CONTENTS.

General—Linear Measures—Square Measures—Cubic Measures—Measures of Capacity—Weights—Combinations—Thermometers.

Elements of Construction for Electro-Magnets. By Count TH. DU MONCEL, Mem. de l'Institut de France. Translated from the French by C. J. WHARTON. Crown 8vo, cloth, 4*s.* 6*d.*

Practical Electrical Units Popularly Explained, with *numerous illustrations* and Remarks. By JAMES SWINBURNE, late of J. W. Swan and Co., Paris, late of Brush-Swan Electric Light Company, U.S.A. 18mo, cloth, 1*s.* 6*d.*

A Treatise on the Use of Belting for the Transmission of Power. By J. H. COOPER. Second edition, *illustrated*, 8vo, cloth, 15*s.*

A Pocket-Book of Useful Formulæ and Memoranda for Civil and Mechanical Engineers. By GUILFORD L. MOLESWORTH, Mem. Inst. C.E., Consulting Engineer to the Government of India for State Railways. *With numerous illustrations*, 744 pp. Twenty-second edition, revised and enlarged, 32mo, roan, 6*s.*

SYNOPSIS OF CONTENTS:

Surveying, Levelling, etc.—Strength and Weight of Materials—Earthwork, Brickwork, Masonry, Arches, etc.—Struts, Columns, Beams, and Trusses—Flooring, Roofing, and Roof Trusses—Girders, Bridges, etc.—Railways and Roads—Hydraulic Formulæ—Canals, Sewers, Waterworks, Docks—Irrigation and Breakwaters—Gas, Ventilation, and Warming—Heat, Light, Colour, and Sound—Gravity: Centres, Forces, and Powers—Millwork, Teeth of Wheels, Shafting, etc.—Workshop Recipes—Sundry Machinery—Animal Power—Steam and the Steam Engine—Water-power, Water-wheels, Turbines, etc.—Wind and Windmills—Steam Navigation, Ship Building, Tonnage, etc.—Gunnery, Projectiles, etc.—Weights, Measures, and Money—Trigonometry, Conic Sections, and Curves—Telegraphy—Mensuration—Tables of Areas and Circumference, and Arcs of Circles—Logarithms, Square and Cube Roots, Powers—Reciprocals, etc.—Useful Numbers—Differential and Integral Calculus—Algebraic Signs—Telegraphic Construction and Formulæ.

Hints on Architectural Draughtsmanship. By G. W. TUXFORD HALLATT. Fcap. 8vo, cloth, 1*s*. 6*d*.

Spons' Tables and Memoranda for Engineers; selected and arranged by J. T. HURST, C.E., Author of 'Architectural Surveyors' Handbook,' 'Hurst's Tredgold's Carpentry,' etc. Ninth edition, 64mo, roan, gilt edges, 1*s*.; or in cloth case, 1*s*. 6*d*.

This work is printed in a pearl type, and is so small, measuring only 2½ in. by 1¾ in. by ⅓ in. thick, that it may be easily carried in the waistcoat pocket.

"It is certainly an extremely rare thing for a reviewer to be called upon to notice a volume measuring but 2½ in. by 1¾ in., yet these dimensions faithfully represent the size of the handy little book before us. The volume—which contains 118 printed pages, besides a few blank pages for memoranda—is, in fact, a true pocket-book, adapted for being carried in the waistcoat pocket, and containing a far greater amount and variety of information than most people would imagine could be compressed into so small a space. . . . . The little volume has been compiled with considerable care and judgment, and we can cordially recommend it to our readers as a useful little pocket companion."—*Engineering*.

A Practical Treatise on Natural and Artificial Concrete, its Varieties and Constructive Adaptations. By HENRY REID, Author of the 'Science and Art of the Manufacture of Portland Cement.' New Edition, *with* 59 *woodcuts and* 5 *plates*, 8vo, cloth, 15*s*.

Notes on Concrete and Works in Concrete; especially written to assist those engaged upon Public Works. By JOHN NEWMAN, Assoc. Mem. Inst. C.E., crown 8vo, cloth, 4*s*. 6*d*.

Electricity as a Motive Power. By Count TH. DU MONCEL, Membre de l'Institut de France, and FRANK GERALDY, Ingénieur des Ponts et Chaussées. Translated and Edited, with Additions, by C. J. WHARTON, Assoc. Soc. Tel. Eng. and Elec. *With* 113 *engravings and diagrams*, crown 8vo, cloth, 7*s*. 6*d*.

Treatise on Valve-Gears, with special consideration of the Link-Motions of Locomotive Engines. By Dr. GUSTAV ZEUNER, Professor of Applied Mechanics at the Confederated Polytechnikum of Zurich. Translated from the Fourth German Edition, by Professor J. F. KLEIN, Lehigh University, Bethlehem, Pa. *Illustrated*, 8vo, cloth, 12*s*. 6*d*.

The French-Polisher's Manual. By a French-Polisher; containing Timber Staining, Washing, Matching, Improving, Painting, Imitations, Directions for Staining, Sizing, Embodying, Smoothing, Spirit Varnishing, French-Polishing, Directions for Re-polishing. Third edition, royal 32mo, sewed, 6*d*.

Hops, their Cultivation, Commerce, and Uses in various Countries. By P. L. SIMMONDS. Crown 8vo, cloth, 4*s*. 6*d*.

The Principles of Graphic Statics. By GEORGE SYDENHAM CLARKE, Capt. Royal Engineers. *With* 112 *illustrations*. 4to, cloth, 12*s*. 6*d*.

Dynamo-Electric Machinery: A Manual for Students of Electro-technics. By SILVANUS P. THOMPSON, B.A., D.Sc., Professor of Experimental Physics in University College, Bristol, etc., etc. Third edition, *illustrated*, 8vo, cloth, 16*s.*

Practical Geometry, Perspective, and Engineering Drawing; a Course of Descriptive Geometry adapted to the Requirements of the Engineering Draughtsman, including the determination of cast shadows and Isometric Projection, each chapter being followed by numerous examples; to which are added rules for Shading, Shade-lining, etc., together with practical instructions as to the Lining, Colouring, Printing, and general treatment of Engineering Drawings, with a chapter on drawing Instruments. By GEORGE S. CLARKE, Capt. R.E. Second edition, *with* 21 *plates.* 2 vols., cloth, 10*s.* 6*d.*

The Elements of Graphic Statics. By Professor KARL VON OTT, translated from the German by G. S. CLARKE, Capt. R.E., Instructor in Mechanical Drawing, Royal Indian Engineering College. *With* 93 *illustrations*, crown 8vo, cloth, 5*s.*

A Practical Treatise on the Manufacture and Distribution of Coal Gas. By WILLIAM RICHARDS. Demy 4to, with *numerous wood engravings and* 29 *plates*, cloth, 28*s.*

SYNOPSIS OF CONTENTS:

Introduction — History of Gas Lighting — Chemistry of Gas Manufacture, by Lewis Thompson, Esq., M.R.C.S.—Coal, with Analyses, by J. Paterson, Lewis Thompson, and G. R. Hislop, Esqrs.—Retorts, Iron and Clay—Retort Setting—Hydraulic Main—Condensers — Exhausters — Washers and Scrubbers — Purifiers — Purification — History of Gas Holder — Tanks, Brick and Stone, Composite, Concrete, Cast-iron, Compound Annular Wrought-iron — Specifications — Gas Holders — Station Meter — Governor — Distribution— Mains—Gas Mathematics, or Formulæ for the Distribution of Gas, by Lewis Thompson, Esq.— Services—Consumers' Meters—Regulators—Burners—Fittings—Photometer—Carburization of Gas—Air Gas and Water Gas—Composition of Coal Gas, by Lewis Thompson, Esq.— Analyses of Gas—Influence of Atmospheric Pressure and Temperature on Gas—Residual Products—Appendix—Description of Retort Settings, Buildings, etc., etc.

The New Formula for Mean Velocity of Discharge of Rivers and Canals. By W. R. KUTTER. Translated from articles in the 'Cultur-Ingénieur,' by LOWIS D'A. JACKSON, Assoc. Inst. C.E. 8vo, cloth, 12*s.* 6*d.*

The Practical Millwright and Engineer's Ready Reckoner; or Tables for finding the diameter and power of cog-wheels, diameter, weight, and power of shafts, diameter and strength of bolts, etc. By THOMAS DIXON. Fourth edition, 12mo, cloth, 3*s.*

Tin: Describing the Chief Methods of Mining, Dressing and Smelting it abroad; with Notes upon Arsenic, Bismuth and Wolfram. By ARTHUR G. CHARLETON, Mem. American Inst. of Mining Engineers. *With plates*, 8vo, cloth, 12*s.* 6*d.*

Perspective, Explained and Illustrated. By G. S. CLARKE, Capt. R.E. *With illustrations*, 8vo, cloth, 3*s*. 6*d*.

Practical Hydraulics; a Series of Rules and Tables for the use of Engineers, etc., etc. By THOMAS BOX. Fifth edition, *numerous plates*, post 8vo, cloth, 5*s*.

The Essential Elements of Practical Mechanics; based on the Principle of Work, designed for Engineering Students. By OLIVER BYRNE, formerly Professor of Mathematics, College for Civil Engineers. Third edition, *with* 148 *wood engravings*, post 8vo, cloth, 7*s*. 6*d*.

CONTENTS:

Chap. 1. How Work is Measured by a Unit, both with and without reference to a Unit of Time—Chap. 2. The Work of Living Agents, the Influence of Friction, and introduces one of the most beautiful Laws of Motion—Chap. 3. The principles expounded in the first and second chapters are applied to the Motion of Bodies—Chap. 4. The Transmission of Work by simple Machines—Chap. 5. Useful Propositions and Rules.

Breweries and Maltings: their Arrangement, Construction, Machinery, and Plant. By G. SCAMELL, F.R.I.B.A. Second edition, revised, enlarged, and partly rewritten. By F. COLYER, M.I.C.E., M.I.M.E. *With* 20 *plates*, 8vo, cloth, 18*s*.

A Practical Treatise on the Construction of Horizontal and Vertical Waterwheels, specially designed for the use of operative mechanics. By WILLIAM CULLEN, Millwright and Engineer. *With* 11 *plates*. Second edition, revised and enlarged, small 4to, cloth, 12*s*. 6*d*.

A Practical Treatise on Mill-gearing, Wheels, Shafts, Riggers, etc.; for the use of Engineers. By THOMAS BOX. Third edition, *with* 11 *plates*. Crown 8vo, cloth, 7*s*. 6*d*.

Mining Machinery: a Descriptive Treatise on the Machinery, Tools, and other Appliances used in Mining. By G. G. ANDRÉ, F.G.S., Assoc. Inst. C.E., Mem. of the Society of Engineers. Royal 4to, uniform with the Author's Treatise on Coal Mining, containing 182 *plates*, accurately drawn to scale, with descriptive text, in 2 vols., cloth, 3*l*. 12*s*.

CONTENTS:

Machinery for Prospecting, Excavating, Hauling, and Hoisting—Ventilation—Pumping—Treatment of Mineral Products, including Gold and Silver, Copper, Tin, and Lead, Iron Coal, Sulphur, China Clay, Brick Earth, etc.

Tables for Setting out Curves for Railways, Canals, Roads, etc., varying from a radius of five chains to three miles. By A. KENNEDY and R. W. HACKWOOD. *Illustrated*, 32mo, cloth, 2*s*. 6*d*.

The Science and Art of the Manufacture of Portland Cement, with observations on some of its constructive applications. *With 66 illustrations.* By HENRY REID, C.E., Author of 'A Practical Treatise on Concrete,' etc., etc. 8vo, cloth, 18*s.*

The Draughtsman's Handbook of Plan and Map Drawing; including instructions for the preparation of Engineering, Architectural, and Mechanical Drawings. *With numerous illustrations in the text, and* 33 *plates* (15 *printed in colours*). By G. G. ANDRÉ, F.G.S., Assoc. Inst. C.E. 4to, cloth, 9*s.*

CONTENTS:

The Drawing Office and its Furnishings—Geometrical Problems—Lines, Dots, and their Combinations—Colours, Shading, Lettering, Bordering, and North Points—Scales—Plotting—Civil Engineers' and Surveyors' Plans—Map Drawing—Mechanical and Architectural Drawing—Copying and Reducing Trigonometrical Formulæ, etc., etc.

The Boiler-maker's and Iron Ship-builder's Companion, comprising a series of original and carefully calculated tables, of the utmost utility to persons interested in the iron trades. By JAMES FODEN, author of 'Mechanical Tables,' etc. Second edition revised, *with illustrations,* crown 8vo, cloth, 5*s.*

Rock Blasting: a Practical Treatise on the means employed in Blasting Rocks for Industrial Purposes. By G. G. ANDRÉ, F.G.S., Assoc. Inst. C.E. *With* 56 *illustrations and* 12 *plates,* 8vo, cloth, 10*s.* 6*d.*

Painting and Painters' Manual: a Book of Facts for Painters and those who Use or Deal in Paint Materials. By C. L. CONDIT and J. SCHELLER. *Illustrated,* 8vo, cloth, 10*s.* 6*d.*

A Treatise on Ropemaking as practised in public and private Rope-yards, with a Description of the Manufacture, Rules, Tables of Weights, etc., adapted to the Trade, Shipping, Mining, Railways, Builders, etc. By R. CHAPMAN, formerly foreman to Messrs. Huddart and Co., Limehouse, and late Master Ropemaker to H.M. Dockyard, Deptford. Second edition, 12mo, cloth, 3*s.*

Laxton's Builders' and Contractors' Tables; for the use of Engineers, Architects, Surveyors, Builders, Land Agents, and others. Bricklayer, containing 22 tables, with nearly 30,000 calculations. 4to, cloth, 5*s.*

Laxton's Builders' and Contractors' Tables. Excavator, Earth, Land, Water, and Gas, containing 53 tables, with nearly 24,000 calculations. 4to, cloth, 5*s.*

Sanitary Engineering: a Guide to the Construction of Works of Sewerage and House Drainage, with Tables for facilitating the calculations of the Engineer. By BALDWIN LATHAM, C.E., M. Inst. C.E., F.G.S., F.M.S., Past-President of the Society of Engineers. Second edition, *with numerous plates and woodcuts*, 8vo, cloth, 1*l.* 10*s.*

Screw Cutting Tables for Engineers and Machinists, giving the values of the different trains of Wheels required to produce Screws of any pitch, calculated by Lord Lindsay, M.P., F.R.S., F.R.A.S., etc. Cloth, oblong, 2*s.*

Screw Cutting Tables, for the use of Mechanical Engineers, showing the proper arrangement of Wheels for cutting the Threads of Screws of any required pitch, with a Table for making the Universal Gas-pipe Threads and Taps. By W. A. MARTIN, Engineer. Second edition, oblong, cloth, 1*s.*, or sewed, 6*d.*

A Treatise on a Practical Method of Designing Slide-Valve Gears by Simple Geometrical Construction, based upon the principles enunciated in Euclid's Elements, and comprising the various forms of Plain Slide-Valve and Expansion Gearing; together with Stephenson's, Gooch's, and Allan's Link-Motions, as applied either to reversing or to variable expansion combinations. By EDWARD J. COWLING WELCH, Memb. Inst. Mechanical Engineers. Crown 8vo, cloth, 6*s.*

Cleaning and Scouring: a Manual for Dyers, Laundresses, and for Domestic Use. By S. CHRISTOPHER. 18mo, sewed, 6*d.*

A Glossary of Terms used in Coal Mining. By WILLIAM STUKELEY GRESLEY, Assoc. Mem. Inst. C.E., F.G.S., Member of the North of England Institute of Mining Engineers. *Illustrated with numerous woodcuts and diagrams*, crown 8vo, cloth, 5*s.*

A Pocket-Book for Boiler Makers and Steam Users, comprising a variety of useful information for Employer and Workman, Government Inspectors, Board of Trade Surveyors, Engineers in charge of Works and Slips, Foremen of Manufactories, and the general Steam-using Public. By MAURICE JOHN SEXTON. Second edition, royal 32mo, roan, gilt edges, 5*s.*

Electrolysis: a Practical Treatise on Nickeling, Coppering, Gilding, Silvering, the Refining of Metals, and the treatment of Ores by means of Electricity. By HIPPOLYTE FONTAINE, translated from the French by J. A. BERLY, C.E., Assoc. S.T.E. *With engravings.* 8vo, cloth, 9*s.*

Barlow's Tables of Squares, Cubes, Square Roots, Cube Roots, Reciprocals of all Integer Numbers up to 10,000. Post 8vo, cloth, 6*s*.

A Practical Treatise on the Steam Engine, containing Plans and Arrangements of Details for Fixed Steam Engines, with Essays on the Principles involved in Design and Construction. By ARTHUR RIGG, Engineer, Member of the Society of Engineers and of the Royal Institution of Great Britain. Demy 4to, *copiously illustrated with woodcuts and* 96 *plates*, in one Volume, half-bound morocco, 2*l*. 2*s*.; or cheaper edition, cloth, 25*s*.

This work is not, in any sense, an elementary treatise, or history of the steam engine, but is intended to describe examples of Fixed Steam Engines without entering into the wide domain of locomotive or marine practice. To this end illustrations will be given of the most recent arrangements of Horizontal, Vertical, Beam, Pumping, Winding, Portable, Semi-portable, Corliss, Allen, Compound, and other similar Engines, by the most eminent Firms in Great Britain and America. The laws relating to the action and precautions to be observed in the construction of the various details, such as Cylinders, Pistons, Piston-rods, Connecting-rods, Cross-heads, Motion-blocks, Eccentrics, Simple, Expansion, Balanced, and Equilibrium Slide-valves, and Valve-gearing will be minutely dealt with. In this connection will be found articles upon the Velocity of Reciprocating Parts and the Mode of Applying the Indicator, Heat and Expansion of Steam Governors, and the like. It is the writer's desire to draw illustrations from every possible source, and give only those rules that present practice deems correct.

A Practical Treatise on the Science of Land and Engineering Surveying, Levelling, Estimating Quantities, etc., with a general description of the several Instruments required for Surveying, Levelling, Plotting, etc. By H. S. MERRETT. Fourth edition, revised by G. W. USILL, Assoc. Mem. Inst. C.E. 41 *plates, with illustrations and tables*, royal 8vo, cloth, 12*s*. 6*d*.

PRINCIPAL CONTENTS:

Part 1. Introduction and the Principles of Geometry. Part 2. Land Surveying; comprising General Observations—The Chain—Offsets Surveying by the Chain only—Surveying Hilly Ground—To Survey an Estate or Parish by the Chain only—Surveying with the Theodolite—Mining and Town Surveying—Railroad Surveying—Mapping—Division and Laying out of Land—Observations on Enclosures—Plane Trigonometry. Part 3. Levelling—Simple and Compound Levelling—The Level Book—Parliamentary Plan and Section—Levelling with a Theodolite—Gradients—Wooden Curves—To Lay out a Railway Curve—Setting out Widths. Part 4. Calculating Quantities generally for Estimates—Cuttings and Embankments—Tunnels—Brickwork—Ironwork—Timber Measuring. Part 5. Description and Use of Instruments in Surveying and Plotting—The Improved Dumpy Level—Troughton's Level—The Prismatic Compass—Proportional Compass—Box Sextant—Vernier—Pantagraph—Merrett's Improved Quadrant—Improved Computation Scale—The Diagonal Scale—Straight Edge and Sector. Part 6. Logarithms of Numbers—Logarithmic Sines and Co-Sines, Tangents and Co-Tangents—Natural Sines and Co-Sines—Tables for Earthwork, for Setting out Curves, and for various Calculations, etc., etc., etc.

Health and Comfort in House Building, or Ventilation with Warm Air by Self-Acting Suction Power, with Review of the mode of Calculating the Draught in Hot-Air Flues, and with some actual Experiments. By J. DRYSDALE, M.D., and J. W. HAYWARD, M.D. Second edition, with Supplement, *with plates*, demy 8vo, cloth, 7*s*. 6*d*.

The Assayer's Manual: an Abridged Treatise on the Docimastic Examination of Ores and Furnace and other Artificial Products. By BRUNO KERL. Translated by W. T. BRANNT. *With* 65 *illustrations*, 8vo, cloth, 12*s*. 6*d*.

Electricity: its Theory, Sources, and Applications. By J. T. SPRAGUE, M.S.T.E. Second edition, revised and enlarged, *with numerous illustrations*, crown 8vo, cloth, 15*s*.

The Practice of Hand Turning in Wood, Ivory, Shell, etc., with Instructions for Turning such Work in Metal as may be required in the Practice of Turning in Wood, Ivory, etc.; also an Appendix on Ornamental Turning. (A book for beginners.) By FRANCIS CAMPIN. Third edition, *with wood engravings*, crown 8vo, cloth, 6*s*.

CONTENTS:

On Lathes—Turning Tools—Turning Wood—Drilling—Screw Cutting—Miscellaneous Apparatus and Processes—Turning Particular Forms—Staining—Polishing—Spinning Metals—Materials—Ornamental Turning, etc.

Treatise on Watchwork, Past and Present. By the Rev. H. L. NELTHROPP, M.A., F.S.A. *With* 32 *illustrations*, crown 8vo, cloth, 6*s*. 6*d*.

CONTENTS:

Definitions of Words and Terms used in Watchwork—Tools—Time—Historical Summary—On Calculations of the Numbers for Wheels and Pinions; their Proportional Sizes, Trains, etc.—Of Dial Wheels, or Motion Work—Length of Time of Going without Winding up—The Verge—The Horizontal—The Duplex—The Lever—The Chronometer—Repeating Watches—Keyless Watches—The Pendulum, or Spiral Spring—Compensation—Jewelling of Pivot Holes—Clerkenwell—Fallacies of the Trade—Incapacity of Workmen—How to Choose and Use a Watch, etc.

Algebra Self-Taught. By W. P. HIGGS, M.A., D.Sc., LL.D., Assoc. Inst. C.E., Author of 'A Handbook of the Differential Calculus,' etc. Second edition, crown 8vo, cloth, 2*s*. 6*d*.

CONTENTS:

Symbols and the Signs of Operation—The Equation and the Unknown Quantity—Positive and Negative Quantities—Multiplication—Involution—Exponents—Negative Exponents—Roots, and the Use of Exponents as Logarithms—Logarithms—Tables of Logarithms and Proportionate Parts—Transformation of System of Logarithms—Common Uses of Common Logarithms—Compound Multiplication and the Binomial Theorem—Division, Fractions, and Ratio—Continued Proportion—The Series and the Summation of the Series—Limit of Series—Square and Cube Roots—Equations—List of Formulæ, etc.

Spons' Dictionary of Engineering, Civil, Mechanical, Military, and Naval; with technical terms in French, German, Italian, and Spanish, 3100 pp., and *nearly* 8000 *engravings*, in super-royal 8vo, in 8 divisions, 5*l*. 8*s*. Complete in 3 vols., cloth, 5*l*. 5*s*. Bound in a superior manner, half-morocco, top edge gilt, 3 vols., 6*l*. 12*s*.

Notes in Mechanical Engineering. Compiled principally for the use of the Students attending the Classes on this subject at the City of London College. By HENRY ADAMS, Mem. Inst. M.E., Mem. Inst. C.E., Mem. Soc. of Engineers. Crown 8vo, cloth, 2*s.* 6*d.*

Canoe and Boat Building: a complete Manual for Amateurs, containing plain and comprehensive directions for the construction of Canoes, Rowing and Sailing Boats, and Hunting Craft. By W. P. STEPHENS. *With numerous illustrations and 24 plates of Working Drawings.* Crown 8vo, cloth, 7*s.* 6*d.*

Proceedings of the National Conference of Electricians, Philadelphia, October 8th to 13th, 1884. 18mo, cloth, 3*s.*

Dynamo - Electricity, its Generation, Application, Transmission, Storage, and Measurement. By G. B. PRESCOTT. *With 545 illustrations.* 8vo, cloth, 1*l.* 1*s.*

Domestic Electricity for Amateurs. Translated from the French of E. HOSPITALIER, Editor of "L'Electricien," by C. J. WHARTON, Assoc. Soc. Tel. Eng. *Numerous illustrations.* Demy 8vo, cloth, 9*s.*

CONTENTS:

1. Production of the Electric Current—2. Electric Bells—3. Automatic Alarms—4. Domestic Telephones—5. Electric Clocks—6. Electric Lighters—7. Domestic Electric Lighting—8. Domestic Application of the Electric Light—9. Electric Motors—10. Electrical Locomotion—11. Electrotyping, Plating, and Gilding—12. Electric Recreations—13. Various applications—Workshop of the Electrician.

Wrinkles in Electric Lighting. By VINCENT STEPHEN. *With illustrations.* 18mo, cloth, 2*s.* 6*d.*

CONTENTS:

1. The Electric Current and its production by Chemical means—2. Production of Electric Currents by Mechanical means—3. Dynamo-Electric Machines—4. Electric Lamps—5. Lead—6. Ship Lighting.

The Practical Flax Spinner; being a Description of the Growth, Manipulation, and Spinning of Flax and Tow. By LESLIE C. MARSHALL, of Belfast. *With illustrations.* 8vo, cloth, 15*s.*

Foundations and Foundation Walls for all classes of Buildings, Pile Driving, Building Stones and Bricks, Pier and Wall construction, Mortars, Limes, Cements, Concretes, Stuccos, &c. 64 *illustrations.* By G. T. POWELL and F. BAUMAN. 8vo, cloth, 10*s.* 6*d.*

Manual for Gas Engineering Students. By D. LEE. 18mo, cloth 1*s*.

Hydraulic Machinery, Past and Present. A Lecture delivered to the London and Suburban Railway Officials' Association. By H. ADAMS, Mem. Inst. C.E. *Folding plate.* 8vo, sewed, 1*s*.

Twenty Years with the Indicator. By THOMAS PRAY, Jun., C.E., M.E., Member of the American Society of Civil Engineers. 2 vols., royal 8vo, cloth, 12*s*. 6*d*.

Annual Statistical Report of the Secretary to the Members of the Iron and Steel Association on the Home and Foreign Iron and Steel Industries in 1884. Issued March 1885. 8vo, sewed, 5*s*.

Bad Drains, and How to Test them; with Notes on the Ventilation of Sewers, Drains, and Sanitary Fittings, and the Origin and Transmission of Zymotic Disease. By R. HARRIS REEVES. Crown 8vo, cloth, 3*s*. 6*d*.

Well Sinking. The modern practice of Sinking and Boring Wells, with geological considerations and examples of Wells. By ERNEST SPON, Assoc. Mem. Inst. C.E., Mem. Soc. Eng., and of the Franklin Inst., etc. Second edition, revised and enlarged. Crown 8vo, cloth, 10*s*. 6*d*.

Pneumatic Transmission of Messages and Parcels between Paris and London, viâ Calais and Dover. By J. B. BERLIER, C.E. Small folio, sewed, 6*d*.

List of Tests (Reagents), arranged in alphabetical order, according to the names of the originators. Designed especially for the convenient reference of Chemists, Pharmacists, and Scientists. By HANS M. WILDER. Crown 8vo, cloth, 4*s*. 6*d*.

Ten Years' Experience in Works of Intermittent Downward Filtration. By J. BAILEY DENTON, Mem. Inst. C.E. Second edition, with additions. Royal 8vo, sewed, 4*s*.

A Treatise on the Manufacture of Soap and Candles, Lubricants and Glycerin. By W. LANT CARPENTER, B.A., B.Sc. (late of Messrs. C. Thomas and Brothers, Bristol). *With illustrations.* Crown 8vo, cloth, 10*s*. 6*d*.

The Stability of Ships explained simply, and calculated by a new Graphic method. By J. C. SPENCE, M.I.N.A. 4to, sewed, 3*s.* 6*d.*

Steam Making, or Boiler Practice. By CHARLES A. SMITH, C.E. 8vo, cloth, 10*s.* 6*d.*

CONTENTS:

1. The Nature of Heat and the Properties of Steam—2. Combustion.—3. Externally Fired Stationary Boilers—4. Internally Fired Stationary Boilers—5. Internally Fired Portable Locomotive and Marine Boilers—6. Design, Construction, and Strength of Boilers—7. Proportions of Heating Surface, Economic Evaporation, Explosions—8. Miscellaneous Boilers, Choice of Boiler Fittings and Appurtenances.

The Fireman's Guide; a Handbook on the Care of Boilers. By TEKNOLOG, föreningen T. I. Stockholm. Translated from the third edition, and revised by KARL P. DAHLSTROM, M.E. Second edition. Fcap. 8vo, cloth, 2*s.*

A Treatise on Modern Steam Engines and Boilers, including Land Locomotive, and Marine Engines and Boilers, for the use of Students. By FREDERICK COLYER, M. Inst. C.E., Mem. Inst. M.E. *With* 36 *plates.* 4to, cloth, 25*s.*

CONTENTS:

1. Introduction—2. Original Engines—3. Boilers—4. High-Pressure Beam Engines—5. Cornish Beam Engines—6. Horizontal Engines—7. Oscillating Engines—8. Vertical High-Pressure Engines—9. Special Engines—10. Portable Engines—11. Locomotive Engines—12. Marine Engines.

Steam Engine Management; a Treatise on the Working and Management of Steam Boilers. By F. COLYER, M. Inst. C.E., Mem. Inst. M.E. 18mo, cloth, 2*s.*

Land Surveying on the Meridian and Perpendicular System. By WILLIAM PENMAN, C.E. 8vo, cloth, 8*s.* 6*d.*

The Topographer, his Instruments and Methods, designed for the use of Students, Amateur Photographers, Surveyors, Engineers, and all persons interested in the location and construction of works based upon Topography. *Illustrated with numerous plates, maps, and engravings.* By LEWIS M. HAUPT, A.M. 8vo, cloth, 18*s.*

A Text-Book of Tanning, embracing the Preparation of all kinds of Leather. By HARRY R. PROCTOR, F.C.S., of Low Lights Tanneries. *With illustrations.* Crown 8vo, cloth, 10*s.* 6*d.*